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THE ANNUAL MEETING

Since the close of last winter's season the Annual Meeting, to be held in New York, December 3-6, has been receiving the attention of the Committee on Meetings and sessions have been definitely laid out and arrangements made for the solicitation and examination of papers by sub-committees appointed for that purpose. As already announced, a joint meeting of these sub-committees and of the Committee on Meetings was held in the rooms of the Society last July to discuss plans for the Annual Meeting and to determine which of the committees would be prepared to take a session at that time for the presentation of subjects coming within its scope. There will be sessions held in charge of the following sub-committees: textiles, administration, cement manufacture, machine shop practice, and railroads, and the Gas Power Section will also hold its usual meeting. Other sub-committees also may contribute one or more papers to which parts of sessions will be devoted and there will also be contributions by local sections of papers which have been presented with marked success at monthly meetings in different cities. For so early a date, an unusually large number of papers are already on hand, some of which appear in this and in the September issues of The Journal.

Recently the various committees interested in the entertainment of members and guests at the time of the Anuual Meeting have held a conference to discuss ways and means for meeting the expenses of such entertainment. Heretofore these expenses have been met mainly by subscriptions solicited from the New York membership. It is now proposed that certain of the functions, such as the President's Reception which is distinctly a Society affair, shall be paid for out of the general funds, but that it will be in accord with the wishes of the membership at large and with what is coming to be the accepted procedure among the leading organizations, if each member who attends shall bear his proportion of the expenses for the social affairs not strictly related to the work of the Society, such, for example, as the luncheons between sessions and the usual reception and dance on Thursday evening of the Convention. Circulars are now being sent to the membership outlining the plan more in detail and calling for a statement from those who are to attend as to what entertainment will be desired.

Altogether the indications are that the coming Annual Meeting will be among the very best and perhaps the most elaborate which has been held in point of professional meetings and that the entertainment features will be conducted on a better basis.

On Wednesday evening of the meeting there will be a feature in which the entire membership will wish to participate, the reception which the members will tender to Prof. John E. Sweet, Honorary Member and Past-President, to whose work in the formation of the Society all are so greatly indebted.

MEETING IN GERMANY

Arrangements for the meeting of the Society with the Verein deutscher Ingenieure, which promises to be the most remarkable tour of the industries in Germany ever offered to American engineers, are progressing satisfactorily. Mr. Conrad Matschoss, Dozent of the Royal Polytechnic High School of Berlin, is now visiting the United States as a representative of the German society and conferring with the Committee of Arrangements regarding itinerary, professional papers, etc. On the evening of October 8 a dinner will be tendered to Mr. Matschoss by the Council.

In general, the date of departure will probably be June 11,

making it possible for the party to arrive in Germany on Saturday, June 21. Here the party will be received by the municipal authorities and given an opportunity to visit the great shipyards, before going on by special train to Leipzig, either on the same or the following day. Hotel reservations in Leipzig have already been made, so that the party will be ensured of satisfactory accommodations. Sunday evening, the 22d, will be the occasion of an informal gathering to promote acquaintanceship, preceding the more formal opening on Monday morning, June 23. At this opening there will be a formal welcome by the King of Saxony and the municipal authorities, and two addresses, one by a representative of the Verein and the other by a representative of our Society, which will probably be on general subjects, such as the history of engineering, the relation of capital and labor, and the effect on industry of technical education. Tuesday, the last day to be spent in Leipzig, will be devoted to technical papers and to various social events and participation in the patriotic gathering in commemoration of the battle of Leipzig, followed by a gala concert by the Gewandhaus Orchestra under Mr. Arthur Nikisch.

The party will leave on Wednesday for a tour of industrial Germany, visiting the principal cities, such as Dresden, Berlin, Essen, where the Baroness von Bohlen (geboren von Krupp) will receive the party at the Krupp works, rarely thrown open to visitors, Dusseldorf and Cologne, with receptions by municipal authorities, and will make a trip up the Rhine as far as Frankfort. At Rudesheim there will be a grand festival on the embankments, and at Mannheim the Hoeschst Chemical Works will be visited. On July 4 the members of the Society will participate in the customary American celebration arranged by the Ambassador to Germany.

The final meeting will be held in Munich about July 10, in connection with the Museum of Technical Arts, of which Dr. von Miller is director. It is expected that the Prince of Bavaria and the Mayor of Munich will also receive the party. The entire trip through Germany and the various official receptions are being arranged by a committee of the German society and there is already the keenest interest shown in the visit by municipal authorities and the heads of industrial establishments.

To assist our Committee of Arrangements it is essential that

all those who expect to make the trip shall give their decision at once so that the best accommodations may be secured. The larger the party, the better will be the facilities which the transportation companies will offer. Guests may be included in the trip.

CURRENT AFFAIRS OF THE SOCIETY

The Secretary is able to announce that the subscriptions to the Kelvin Memorial which have thus far been received amount to $510, one hundred and forty members having contributed, and that as soon as complete returns are received the entire fund will be sent to the committee in England, together with the amounts secured by the American Society of Civil Engineers, the American Institute of Mining Engineers, the American Institute of Electrical Engineers, the Canadian Society of Civil Engineers, and the Society of Naval Architects and Marine Engineers. This memorial is a fitting tribute to one of the greatest members of the engineering profession and is one in which every engineer of the English-speaking world should have a part.

The Society history which has been in preparation by the committee appointed by the Council will probably be completed this fall and published in the spring of 1913. The members of the committee are John E. Sweet, H. H. Suplee, and F. R. Hutton, Professor Hutton having been appointed to the place of the late Charles Wallace Hunt.

On August 21, many prominent engineers and business men joined in the congratulations sent to John Fritz, Honorary Member and Past-President of the Society, on his ninetieth birthday. The good wishes of the Society were conveyed to Mr. Fritz by the Secretary in a telegram.

CALVIN W. RICE, *Secretary.*

CORRECTION IN SEPTEMBER ISSUE OF THE JOURNAL

Attention is drawn to the following error, occurring on page 1370, line 11, of the September issue of The Journal, in the closure of A. G. Christie: "the efficiency ratio of the unit shall not exceed that represented by the guarantee," should read, "the efficiency ratio of the unit shall not *be less than* that represented by the guarantee."

NEW HAVEN MEETING

The fall meeting of the New Haven Section will be held on Wednesday, November 13, with afternoon and evening sessions, at the Mason Laboratory of the Sheffield Scientific School. In the announcement of this meeting issued by the Committee on Meetings for New Haven, the following four subjects are suggested for future meetings upon which members in the vicinity are asked to express their choice and to offer papers for discussion: the market for Connecticut manufactured products; garbage disposal for cities; the transportation of materials in manufacturing plants; and education for engineers.

REPORT OF THE NOMINATING COMMITTEE

The Secretary, according to B 29 of the Constitution and By-Laws, announces the receipt of the following report from the Nominating Committee, consisting of Ambrose Swasey, Chairman, A. E. Cluett, W. B. Gregory, John Hunter, William Schwanhausser: For President, W. F. M. Goss, Urbana, Ill.; Vice-Presidents (for two years), James Hartness, Springfield, Vt., I. E. Moultrop, Boston, Mass., H. G. Stott, New York; Managers (for three years), W. B. Jackson, Chicago, Ill., H. M. Leland, Detroit, Mich., Alfred Noble, New York; Treasurer, Wm. H. Wiley. All of the nominees have been notified and have accepted.

TWO INTERNATIONAL CONGRESSES

During the past month two international congresses have been held in this country: the Sixth Congress of the International Association for Testing Materials and the Eighth International Congress of Applied Chemistry.

THE INTERNATIONAL ASSOCIATION FOR TESTING MATERIALS

The International Association for Testing Materials met in the Engineering Societies Building, New York, September 2 to 7, with an attendance of more than 700 delegates from 28 different countries. On the evening of the opening day a reception was tendered the delegates by the Society, in association with the American Society of Testing Materials and the founder societies.

The movement which resulted in the establishment of this association was started in 1882 by J. Bauschinger, for the purpose of establishing uniform methods of testing the most important materials of construction. Four conferences were held under the chairmanship of the founder: at Munich in 1884, at Dresden in 1886, at Berlin in 1890 and at Vienna in 1893. At first only nations speaking the German language were represented, but the conferences rapidly assumed an international character. In 1893 the United States was represented by Gustavus C. Henning, Mem.Am.Soc.M.E.

The first congress met in Zurich in 1895, L. Tetmajer, president, and it was here decided to replace the somewhat informal meetings by a permanent organization under the name of the International Association for Testing Materials. In the same year was started the publication of the official organ of the association, Die Baumaterialienkunde. The second congress took place in Stockholm in 1897 when the association was joined by English members of the Iron and Steel Institute. Such papers as Osmond's Metallography as a Method of Testing, Rejto's Internal Friction, and Carnot's Chemical Analysis of Iron, presented before that meeting have a definitely recognized place in the history of engineering. In 1901 when the association held its

third congress in Budapest, the membership had grown to 1800. Two papers made this meeting a notable one, Considère's Properties of Reinforced Concrete, which was then only slightly known as a material of construction, and Baron Jornstorff's, which discussed the forms of carbon in iron and showed that the chemical analysis of metals as carried on up to that time was absolutely inadequate, and that a rational method of analysis was required to determine the forms and combinations of the constitutent elements of metals.

For the fourth congress in Brussels in 1906 Le Chatelier and Guillet organized a laboratory containing some of the most modern testing machinery for both mechanical and metallographic tests of metals. Many of the papers presented before the meeting were the results of the work of international commissions. This is an important feature of the association for it both insures a high quality of the papers submitted and facilitates the adoption of the resolutions of the congresses by official bodies in the various countries.

The fifth congress was held in Copenhagen, with 900 members present, representing 22 countries. Particular attention was paid to metallography: Professor Schule, Dr. Leon and Dr. Ludwik discussed hardness and impact testing, while Dr. Moldenke, Mem.Am.Soc.M.E., presented the official report on uniform methods of testing cast iron. Valuable papers were also read on the testing and properties of cement, stone and concrete. An important branch of the work of the association is the establishment and adoption of international specifications and nomenclature, the report of the committee of the uniform nomenclature of iron and steel, especially the nomenclature of the microscopic constituents of iron and steel, by H. M. Howe and A. Sauveur of this country, being an example.

The association now numbers about 2800 members representing about every country of the world. Among the papers presented at the sixth congress may be mentioned the following: C. Grard (Studies of Hardness of Steels) showed that hardness tests may be used in place of tension tests by multiplying the hardness number by a constant, or may serve for tests of homogeneity, and specified the precautions necessary to attain the required precision; the author considered it practicable to specify hardness and impact test requirements in steel specifications. G. Charpy (committee paper on Impact Tests) stated that the

tests made by the committee with the two sizes of bars approved by the fifth congress (30 by 30 mm., 120 mm. span, and 10 by 10 mm., 40 mm. span) proved that equality of results is not bound to hold, and that smaller bars give smaller values. M. Derihon (Remarks on the Brittleness Test) believed, after having used the Fremont impact test, that was a reliable means of discovering brittleness and was not an unduly severe method of test; the author had used it since 1904, with up to 12,000 tests per month, for all steel bought or treated in his plant, and now had only 0.3 per cent rejections. C. Fremont (A New Testing Method for Steel Tubes) suggested application of the notched bar impact test to tubes and said if a narrow ring of the tube were flattened, doubled to U-shape, notched by a saw cut, and tested in the Fremont machine it should give at least 16 kgm. per sq. cm. (1072 ft-lb. per sq. in.) initial sectional area.

T. E. Stanton gave an account of the recent researches at the National Physical Laboratory at Teddington on the resistance of metals to alternating stresses. He said that rapidity of alterations had no influence on the results up to 7000 per minute, and maximum range of stress was about the same as the high yield-point value; strength for long-time endurance was apt to be in the reverse order of the single blow impact strengths, while repeated impact strength could be computed from the simple fatigue strength. O. Boudouard (Endurance Test of Metals) presented a new method of testing by measuring the damping of vibrations. C. W. Burrows showed that there was a close relationship between the magnetic properties and the presence of flaws, blowholes and strains, while elastic limit, tensile strength, and hardness were, within certain limitations, functions of the magnetic reluctance and electric resistance of the materials. He believed therefore that certain electric and magnetic tests may be used to supplement or replace certain mechanical tests. Several other papers indicated the existence of the same relationship between the mechanical and magnetic properties of steel. F. Robin in his paper, Several Mechanical Properties of Metals in the Hot State, demonstrated that iron and pearlitic steel under static stress showed a minimum strength at about 100 deg., a maximum at about 250 deg., and a bend in the curve at about 500 deg. Under dynamic stress the minimum was found near 300 deg. and the maximum at about 500 deg. cent. The effect of the coldworking of hard metals grew with the temperature and

tended to increase their resistant qualities; on the other hand, heating diminished the resistance progressively, and the two effects would bring about a minimum of the properties of resistance, which accidentally fell near 100 deg. cent. O. Gallander showed how the depth of a crack in a conductive metal may be determined electrically, while A. Mesnager (A Method of Foreseeing Rail Fractures) claimed that surface fissures on the head of a rail have a great influence in causing fractures, and a picture of the surface of a rail with its fissures and defects may help to determine the primary cause of an accident in a case of derailment, by showing whether or not the rail was brittle on account of internal fissures. W. Campbell and J. Glassford found, from a study of 23 specimens of pig, cast iron and steel subjected to prolonged exposure to superheated steam or many repetitions of heating and cooling, that white cast-iron, malleable castings and steel, oxidized superficially, while in gray cast iron the oxidation penetrated deeper, following the graphite flakes, the change increasing with the silicon content. C. Fremont advocated shearing as a method of mechanical testing of cast iron. He proposed the use of a test-piece 5 mm. square worked from a 7.5 mm. cylinder taken out of the casting with a core drill; shearing this test-piece every 3 mm. along its length gave data on strength at different depths below the surface. T. D. West, Mem.Am.Soc.M.E., (A Suggested System of Test-bars for Chillable Irons) stated that in testing chillable irons both the white and the gray phase should be tested, and described the molds for both. R. Moldenke, Mem.Am.Soc.M.E., criticized the tendency to push unduly the quality of a cheap material to its upper limit, rather than to step at once to the higher grade material. De Nolly and Veyret after studying the gases disengaged by various carburizers to determine the cause of explosions which occur in case hardening, concluded that hydrogen emitted at higher temperatures by the case-hardening materials was the probable cause of these explosions. They recommended therefore that carburizers containing much animal or vegetable matter not sufficiently calcined be not used, but instead well calcined charcoal alone or mixed with barium carbonate, the heating being slow up to 700 deg. cent. (1292 deg. fahr.). N. Pecoraro (Researches on White-metals) stated that the temperature of bearing metals seemed to have a distinct influence on their resistance to wear, and the only mechanical tests having practical

value are the hardness test, impact tests which may give indications as to brittleness, and such tests in gradual compression as measure the value of the metal for supporting its working loads. C. Fremont described a new testing method for steel wire, being an adaptation of the impact tests with the wire notched with a file.

In section B, Cement and Stone, several papers were devoted to the determination of the very fine powder in portland cement, but without any definite proposals having been made as yet by the committee. E. G. Coker, by applying the method of polarized light effects in transparent bodies under stress to celluloid models of cement of the English, American, and continental forms established the fact that different test results would be shown in cement tests with briquettes of different forms. J. Y. Jewett in describing the tests of construction materials by the United States Reclamation Service called attention, among other things, to the superiority of sands made from crushed rock, such as granite, limestone and basalt, over natural quartz sands, for the production of concrete, while Bertram Blount (Accelerated Tests for Constancy of Volume of Cement) stated that no case had been found where a cement which stood the Le Chatelier test proved unsound later, and that since this test had been introduced in England there had been no difficulty in obtaining cement that would stand it. Speaking on the basis of experiments made by the Association of German Portland Cement Works, M. Gary stated that comparative tests showed that the Le Chatelier test did not give concordant results in different laboratories, and that the material and make of the apparatus influenced the results; no cement proved unsound in use after passing the standard cold-water test, but the boiling test rejected many cements which proved wholly sound in use. The last statement was not accepted by F. Schule who believed that cements which are to be used in dry exposure should be subjected to the boiling test, as is done in Switzerland.

The association continued its important work on the action of seawater on concrete. V. J. P. de Blocq van Kuffeler reported that experience in Holland indicated that reinforced concrete was sufficiently permanent to warrant its use for marine construction, but the following rules should be strictly adhered to: only slow-setting portland cement should be used; mixing and ramming to be done with great care; trass or other puzzolanic

additions are desirable; the mortar should be dense and the mixture rich; setting in the moist air before immersion in the sea is advantageous. The Porosity of Mortars was the subject of a paper by J. Bied who found that an important degree of impermeability to diffusion could be imparted to mortars exposed to the action of seawater by adding to them at the time of mixing barium aluminate transformed into barium sulphate, the latter, in precipitating, closing the capillary pores; a certain degree of impermeability could also be secured by the addition of fatty or gelatinous materials, such as mineral oils.

In Section C, Miscellaneous Materials, there were presented several papers on the testing of paint materials, treating, among other things, of the influence of the pigment on the underlying metal (A. S. Cushman: Paints for Metallic Structures). G. Janka proposed the inclusion of hardness pressure of wood by the ball-pressure method among the standard timber tests of the association; the hardness of wood was found to correspond generally to the crushing strength, with the exception, however, of trees of eccentric trunk growth where the wide-ringed side was harder than the other, but lower in strength. L. Fea discussed the Practical Determination of the Viscosity of Oils, stating that the Engler figures can be taken as equivalent to absolute measures (it was found that the Engler relative viscosities were linear functions of the Pagliani absolute viscosities which were alike for all the oils), and the wear coefficient could be taken as a linear function of viscosity.

A considerable lack of agreement was evidenced with respect to the adoption of international specifications for materials of construction, both in the section of metals (iron and steel), and in section B, with respect to the standardization of the content of SO_3 in portland cement. Some delegates objected to the adoption of any such specifications partly on principle, because they considered that such an act would involve the International Association in matters lying outside of the domain of the science of engineering; others objected to such specifications, particularly those with regard to portland cement, because their adoption might place at a disadvantage their home industries. As a result, in the case of iron and steel specifications the committee was transformed from a legislative body into one whose work will be the compilation and dissemination of existing specifications in different countries, while the committee on the

content of SO_3 in portland cement was simply instructed to collect more material.

Unlike the former congresses of the association, there were no papers defending the status of metallography of iron and steel. The new science has evidently obtained a recognized position among its older peers, and its status cannot be questioned any more.

Another interesting feature of the congress was the comparatively large number of papers devoted to the testing of materials while in actual use, either by electric or magnetic measurements, or by the investigation, in case of rails, of outside fissures. President Howe in his opening address went so far as to ask, "Who shall say that the pitch of volume or timbre of sound emitted by a rail as a result of a given excitement may not be made to disclose pitilessly its hidden defects and to measure the fitness, not alone of the material of which it is composed, but of the rail as a whole structure? Or, giving rein to our fancy, we hear the inspector report, 'This 100-story building indeed responds to G-sharp, but its timbre has this abnormality and these harmonics are exaggerated.'" The sixth congress was practically the first to bring forward as a practical proposition the testing of metals by indestructible methods.

The next meeting is to be held in St. Petersburg in 1915, with Professor N. Belelubsky, of St. Petersburg, as President.

INTERNATIONAL CONGRESS OF APPLIED CHEMISTRY

The eighth International Congress of Applied Chemistry held in Washington and New York, August 31 to September 11, was attended by about 350 foreign delegates and a total of 2223 persons. The bulk of the papers presented was printed in advance of the meetings and represented the tremendous expenditure of labor incident to the preparation of 24 finely printed volumes. Owing to the difficulty of securing a sufficient number of stenographers for reporting the proceedings in their several languages, an ingenious scheme of reporting the discussions by phonograph was adopted, which, however, did not prove to be an unqualified success.

Although primarily a chemical congress, it contained many papers of considerable interest to the mechanical engineer as well. C. G. Fink reported on the applications of ductile tungsten, a remarkable material, with the melting point higher than that

of platinum, tensile strength higher than that of iron and nickel, practically insoluble in all of the common acids, and so hard that it will readily scratch glass and still be ductile. Another comparatively new metal, manganese bronze, and methods of its testing was the subject of a paper by Wm. M. Corse and V. Skillman who laid particular emphasis on the selection of proper test coupons, and using methods of testing corresponding to the type of coupon selected. The electric furnace naturally attracted much attention. D. A. Lyon pointed out that it was not brought out with the idea of competing with the blast furnace, but for the purpose of developing a furnace and a process which could be utilized where high cost prevents the use of the blast furnace. The commercial operation of the electric furnace depends upon the development of cheap electric power, but outside of water power even the gas engine does not appear at present to be cheap enough to permit of its use for iron reduction furnace work. Bradley Stoughton discussed the still mooted question of the influence of titanium on the strength of iron castings. Only 0.9 lb. per ton of steel is required to give increased resistance to wear, but the treatment must be very carefully carried out if good results are to be obtained.

In the section on cement W. A. Schmidt presented a paper on the Control of Dust in Portland Cement Manufacture by the Cottrell Prescription Process, i.e. by electrostatic means, while W. C. Reibling and F. D. Reyes discussed the physical and chemical properties of portland cement and made recommendations for increasing the efficiency of the present standard cement specifications of the American Society for Testing Materials.

J. C. Olsen and A. E. Ratner presented the results of their experiments on the decomposition of linseed oil during drying, showing that a considerable amount of gases is given off during that process (carbon and hydrogen). Several papers were presented on lubricating oils: T. T. Gray gave the results of comparative tests made with the Engler-Saybolt instruments, with the ratios of the readings of the two instruments given in diagrams, so as to permit a ready conversion from Engler's specific viscosities to Saybolt viscosities and vice versa. H. C. Porter and F. K. Ovitz discussed the question of spontaneous combustion of coal: temperature of the surrounding space is a large factor, and storing of coal in warm places is greatly conducive to spontaneous combustion; the presence of dust and fine coal is also

to be avoided as much as possible. The rate of oxidization of coal is a function of the partial pressure of oxygen present, and therefore of the rate of flow of air, but freshly mined coal absorbs oxygen more readily than coal which has been exposed to the air some time.

The following two statements made by Dr. Karl Duisberg, of Germany, are particularly interesting because of the authority of the speaker. The first refers to the production of cold:

It is hoped in the near future to produce refrigerating machines which, by the use of suitable hydrocarbons, will give temperatures of -80 deg. cent. Plants for the liquefaction of air, producing as low a temperature as -190 deg. cent., are becoming more and more common, and are especially profitable where gas mixtures rich in oxygen, or where pure nitrogen simultaneously produced, can be utilized. The Badische Anilin and Soda Fabrik, in Ludwigshafen on the Rhine, intend to manufacture hydrogen from water gas in a similar way and to utilize the carbon monoxide, which is simultaneously obtained, as a source of power.

Another important statement refers to the commercial production of artificial rubber:

The difficulties which have been overcome were great indeed and those which still remain to be surmounted, in order to produce a substance equal to para caoutchouc in quality and capable of competing with cheap plantation rubber costing only 2 marks per kilo, are still greater. The end in view is that artificial rubber may play as important a role in the markets of the world as does natural rubber. I do not know when synthetic rubber will bring the millions which prophets see in its exploitation, surely not in the immediate future, although synthetic rubber will certainly appear on the market in a very short time. But I hope to live long enough to see Art triumph here also over Nature.

The next Congress of Applied Chemistry will be held in St. Petersburg, Russia, in 1915, Professor Walden, of the Riga Polytechnicum, President.

AMENDMENTS TO CONSTITUTION

RESPECTING ELECTION OF MEMBERS

At the Spring Meeting, Mr. Jesse M. Smith, at the request of the Council, offered the amendments suggested by the Committees on Membership and Increase of Membership.

In general they make the grades of membership similar to the more important American and English societies and simplify the method of election and make it also similar to most societies.

This is accomplished by adding the engineering grade of Associate-Member, intermediate between Member and Junior, and by substituting election by the Council for election by the membership.

Wherever the grade of Associate-Member should be added and the method of electing members should be changed in existing paragraphs of the Constitution, it appears in the subjoined.

The following are the proposed amendments which will be open for discussion at the Annual Meeting in December:

AMENDMENTS TO THE CONSTITUTION TO BE PRESENTED UNDER PROVISION C57

AT THE

ANNUAL MEETING IN NEW YORK, DECEMBER 4, 1912

Present	*Proposed*
C6 The membership of the Society shall consist of Honorary Members, Members, Associates, and Juniors. Honorary Members, Members, and Associates are entitled to vote and to hold office. Juniors shall not be entitled to vote nor to be officers of the Society, but shall be entitled to the other privileges of membership.	C6 The membership of the Society shall consist of Honorary Members, Members, Associates, Associate-Members and Juniors. Honorary Members, Members, Associates and Associate-Members are entitled to vote and to hold office. Juniors are not entitled to vote nor to hold office but are entitled to all the other privileges of membership.
C7 Honorary Members, Members and Associates are entitled to vote on all questions before any meeting of the Society, in person or by proxy, given to a voting member. A proxy shall not be valid for a greater time than for six months.	C7 Honorary-Members, Members, Associates and Associate-Members are entitled to vote on all questions before any meeting of the Society, in person, or by proxy given to a person entitled to vote. A proxy shall not be valid for a greater time than for six months.

C8 Honorary Members shall be persons of acknowledged professional eminence, and their number shall not exceed 25 at any time.

C9 A member shall be thirty years of age or over. He must have been so connected with Engineering as to be competent, as a designer or as a constructor, to take responsible charge of work in his branch of engineering, or he must have served as a teacher of engineering for more than five years.

C10 An Associate shall be thirty years of age or over. He must have been so connected with some branch of engineering or science, or the arts, or industries, that the Council will consider him qualified to coöperate with engineers in the advancement of professional knowledge.

C11 A Junior shall be twenty-one years of age or over. He must have had such engineering experience as will enable him to fill a responsible subordinate position in engineering work, or he must be a graduate of an engineering school. A person who is over thirty years of age shall not be eligible to membership in the Society as a Junior.

C12 The rights and privileges of every Honorary Member, Member, Associate, and Junior shall be personal to himself, and shall not be transferable or transmissible by his own act or by operation of law.

C14 All applications for membership to the grades of Member, Associate, or Junior, shall be presented to the Council, which shall consider and act upon each application, assigning each approved applicant to the grade of membership to which, in the judgment of the Council, his qualifications entitle him. The name of each candidate thus approved by the Council shall, unless objection is made by the applicant, be submitted to the voting membership for election, by means of a letter-ballot.

C15 Associates or Juniors desiring to change their grade of membership shall make application to the Council in the same manner as is required in the case of a new applicant.

C16 Election to membership shall be by sealed letter-ballot as the By-Laws shall provide. Adverse votes to the number of two per cent of the votes cast shall be required to defeat the election of an applicant for any grade of

C8 An Honorary-Member shall be a person of acknowledged professional eminence. The number of Honorary-Members shall not exceed 25 at any time.

C9 A Member shall be an Engineer or Teacher of Applied Science of thirty years of age or over. He must have shown by his experience or by the responsibility of his duties that he is competent to take charge of work in his profession.

C10 An Associate shall be thirty years of age or over. He need not be an Engineer but must have been so connected with some branch of Engineering or science, or the arts, or industries, that the Council will consider him qualified to coöperate with Engineers in the advancement of professional knowledge.

New Article—*insert between* C10 *and* C11

C— An Associate-Member shall be an Engineer or Teacher of Applied Science of twenty-five years of age or over. He must show by his experience or by his duties, that he is competent to execute work in his profession.

C11 A Junior shall be twenty-one years of age or over. He must have had such engineering experience as will enable him to fill a subordinate position in engineering work, or he must be a graduate of an engineering school.

C12 The rights and privileges of every Honorary-Member, Member, Associate, Associate-Member and Junior shall be personal to himself, and shall not be transferable or transmissible by his own act or by operation of law.

C14 All applications for membership to the grades of Member, Associate, Associate-Member or Junior, shall be presented to the Council, which shall consider and act upon each application, assigning each approved applicant to the grade of membership to which, in the judgment of the Council, his qualifications entitle him. The name of each candidate thus approved by the Council shall, unless objection is made by the applicant, be submitted to all the members of the Council for election, by means of a letter-ballot.

C15 Any person desiring to change his grade of membership shall make application to the Council in the same manner as is required in the case of a new applicant.

C16 Election to membership shall be by sealed letter-ballot of the Council as the By-Laws shall provide. Fifteen affirmative votes shall be required for the election of a candidate to membership in any grade. One negative

C16—*Continued*

membership. The Council may, in its discretion, order a second ballot upon a defeated applicant, in which case adverse votes to the number of four per cent of the votes cast shall be required to defeat the election.

C17 The Election of Honorary Members shall be by vote of the Council taken by letter-ballot, as provided in the By-Laws. One dissenting vote shall defeat such election.

C19 The initiation fee for membership in each grade shall be as follows:

For Member	$25.00
For Associate	25.00
For Junior	15.00

C21 The annual dues for membership in each grade shall be as follows:

For Member	$15.00
For Associate	15.00
For Junior	10.00

for the first six years of his membership and thereafter the same as for an Associate.

C22 The Council may, in its discretion, permit any Member or Associate to become a Life Member in the same grade, by the payment at one time of an amount sufficient to purchase from the Equitable Life Assurance Society of New York, an annuity on the life of a person of the age of the applicant equal to the annual dues in his grade. Such Life Member shall not be liable thereafter for annual dues.

C24 Any Member, Associate, or Junior, who shall leave his annual dues unpaid for one year, shall not receive the volume of Transactions until such arrears are paid. Any Member, Associate, or Junior, who shall leave his dues unpaid for two years, shall, in the discretion of the Council, have his name stricken from the roll of membership, and shall cease to have any further rights as such.

C26 The affairs of the Society shall be managed by a Board of Directors chosen from among its Members and Associates, which shall be styled "The Council." The Council shall consist of the President of the Society, who shall be the presiding officer, six Vice-Presidents, nine Managers, the Treasurer, and five Past-Presidents. Five members of the Council shall constitute a quorum for the transaction of business. The Secretary may take part in the deliberations of the Council, but shall not have a vote therein. The Chairman of the Finance Committee shall attend the meetings of the Council and take part in the

vote shall defeat an election to Honorary Membership. Two negative votes shall defeat an election to any other grade.

C17 Strike out altogether present C17.

C19 The initiation fee for membership in each grade shall be as follows:

For Member	$25.00
For Associate	25.00
For Associate-Member	25.00
For Junior	15.00

C21 The annual dues for membership in each grade shall be as follows:

For Member	$15.00
For Associate	15.00
For Associate-Member	15.00
For Junior	10.00

for the first six years of his membership and thereafter the same as for an Associate-Member.

C22 The Council may, in its discretion, permit any Member, Associate or Associate-Member to become a Life-Member in the same grade, by the payment at one time of an amount sufficient to purchase from the Equitable Life Assurance Society of New York, an annuity on the life of a person of the age of the applicant equal to the annual dues in his grade. Such Life-Member shall not be liable thereafter for annual dues.

C24 A Person in any grade of membership who shall leave his annual dues unpaid for one year, shall not receive the volume of Transactions until such arrears are paid. A person who shall leave his dues unpaid for two years, shall, in the discretion of the Council, have his name stricken from the roll of membership, and shall cease to have any further rights as such.

C26 The affairs of the Society shall be managed by a board of directors chosen from among its Members, Associates and Associate Members which shall be styled "The Council." The Council shall consist of the President of the Society, who shall be the presiding officer, six Vice-Presidents, nine Managers, the Treasurer, and the five surviving Past-Presidents who have last held office. Five members of the Council shall constitute a quorum for the transaction of business. The Secretary may take part in the deliberations of the Council, but shall not have a vote therein. The Chairman of the Finance Committee shall attend the meetings of the

C26—*Continued*

discussion of financial questions, but shall not have a vote.

C27 The five surviving Past-Presidents who last held the office shall be the members of the Council, with all the rights, privileges, and duties of the other members of the Council.

C29 Should a vacancy occur in the Council, or in any elective office except the Presidency, through death, resignation or other cause, the Council may elect a Member or Associate to fill the vacancy until the next annual election.

C33 The Council may, by a two-thirds vote of the members present, declare any elective office vacant, on the failure of its incumbent for one year, from inability or otherwise, to attend the Council meetings, or to perform the duties of his office, and shall thereupon appoint a Member or Associate to fill the vacancy until the next Annual Meeting. The said appointment shall not render the appointee ineligible to election to any office.

C34 At each Annual Meeting there shall be elected from among the Members and Associates:

A President, to hold office for one year.

Three Vice-Presidents, each to hold office for two years.

Three Managers, each to hold office for three years.

A Treasurer, to hold office for one year.

C41 The Society shall hold two meetings in each year. The Annual Meeting shall begin in New York City on the first Tuesday in December, and a Semi-Annual Meeting shall be held at such time and place as the Council may appoint. Fifty Members and Associates shall constitute a quorum for the transaction of business.

C42 Special Meetings of the Society may be called at any time at the discretion of the Council, or shall be called by the President upon the written request of fifty members entitled to vote, the notices for such meetings to state the business for which such meeting is called and no other business shall be entertained or transacted at that meeting.

Council and take part in the discussion of financial questions, but shall not have a vote.

C27 Strike out present C27.

C29 Should a vacancy occur in the Council, or in any elective office except the Presidency, through death, resignation or other cause, the Council may elect a Member, Associate or Associate-Member to fill the vacancy until the next annual election.

C33 The Council may, by a two-thirds vote of the members present, declare any elective office vacant, on the failure of its incumbent for one year, from inability or otherwise, to attend the Council meetings, or to perform the duties of his office, and shall thereupon appoint a Member, Associate or Associate-Member to fill the vacancy until the next Annual Meeting. The said appointment shall not render the appointee ineligible to election to any office.

C34 At each Annual Meeting there shall be elected from among the Members, Associates and Associate-Members:

A President, to hold office for one year.

Three Vice-Presidents, each to hold office for two years.

Three Managers, each to hold office for three years.

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Amend C45, by adding at the end of the list of Standing Committees the words: "Committee on Constitution and By-Laws."

C45 The Standing Committees of the Society to be appointed by the President shall be:

Finance Committee.
Committee on Meetings.
Publication Committee.
Membership Committee.
Library Committee.
House Committee.
Research Committee.
Public Relations Committee.
Committee on Constitution and By-Laws.

Present

C48 Special Nominating Committee:

Twenty or more members entitled to vote may constitute themselves a Special Nominating Committee, with the same powers as the Annual Nominating Committee.

C57 At any semi-annual meeting of the Society any member may propose in writing an amendment to this Constitution. Such proposed amendment shall not be voted at that meeting, but shall be open to discussion and to such modification as may be accepted by the proposer. The proposed amendment shall be mailed in printed form by the Secretary to each member of the Society entitled to vote, at least sixty days previous to the next annual meeting, accompanied by comment by the Council, if it so elects. At that annual meeting such proposed amendment shall be presented for discussion and final amendment, and shall subsequently be submitted to all members entitled to vote, provided that twenty votes are cast in favor of such submission. The final vote on adoption shall be by sealed letter-ballot, closing at 12 o'clock noon, on the first Monday of March following.

C58 The letter-ballot, accompanied by the text of the proposed amendment, shall be mailed by the Secretary to each member of the Society entitled to vote, at least thirty days previous to the closure of the voting. The ballots shall be voted, canvassed, and announced as provided in the By-Laws. The adoption of the amendment shall be decided by a majority of the votes cast. An amendment shall take effect on the announcement of its adoption by the Presiding Officer of the semi-annual meeting next following the closure of the vote.

Proposed

C48 Special Nominating Committee:

Twenty or more persons entitled to vote may constitute themselves a Special Nominating Committee, with the same powers as the Annual Nominating Committee.

C57 At any semi-annual meeting of the Society any person entitled to vote may propose in writing an amendment to this Constitution. Such proposed amendment shall not be voted at that meeting, but shall be open to discussion and to such modification as may be accepted by the proposer. The proposed amendment shall be mailed in printed form by the Secretary to each person entitled to vote, at least sixty days previous to the next annual meeting, accompanied by comment by the Council, if it so elects. At that annual meeting such proposed amendment shall be presented for discussion and final amendment, and shall subsequently be submitted to all persons entitled to vote, provided that twenty votes are cast in favor of such submission. The final vote on adoption shall be by sealed letter-ballot, closing at 12 o'clock noon, on the first Monday of March following.

C58 The letter-ballot, accompanied by the text of the proposed amendment, shall be mailed by the Secretary to each person entitled to vote, at least thirty days previous to the closure of the voting. The ballots shall be voted, canvassed, and announced as provided in the By-Laws. The adoption of the amendment shall be decided by a majority of the votes cast. An amendment shall take effect on the announcement of its adoption by the Presiding Officer of the semi-annual meeting next following the closure of the vote.

Renumbering

Renumber Articles of the Constitution after they have been approved by the Society.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for the Bulletin must be in hand before the 12th of the month. The list of men available is made up of members of the Society, and these are on file in the Society office, together with names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

0188 Young engineer, who has specialized in design and manufacture of stokers. Salary $2000. Apply through the Society.

0189 Assistant engineer, permanent position. Must be experienced designer of electric traveling cranes. Accurate and quick at figures; capable of systematizing engineering data. Give references as to experience and personal habits. State technical training and salary desired. Apply through the Society.

0190 New York concern desires first class chief draftsman, capable of handling efficiently a force of about twenty men. Should possess thorough technical knowledge and have special experience in design of centrifugal pumping machinery and steam turbines.

0191 Superintendent wanted for established and growing two-cycle marine motor factory, located in large city in the Middle West. Must be good mechanic, capable of designing jigs, resourceful and a producer. Salary $2000. Apply through the Society.

0192 Head draftsman wanted by firm manufacturing pumping machinery, employing 40 men in engineering department. Must have had a successful experience as an executive, either as chief draftsman or second hand, in a large modern plant, preferably along similar lines of work. A man of sanguine temperament, energetic, enthusiastic, and with initiative, is desired, thoroughly acquainted with up-to-date drawing room methods and system; a good designer and standardizer. Apply through the Society.

0193 Man capable of perfecting not only actual working device but following through the shop and making a success of a folding machine for inserts in connection with newspapers, magazines, etc., capacity of 10,000 an hour. Salary $1500 and upward. Apply through the Society.

0194 Competent engineer to take charge of two boilers of about 30 h.p. and drive ice machines. Must be experienced in foundry work. Salary

$100 per month, with free quarters. Sign agreement for three years. M. Shaik Ipoh, Straits Settlements, British India.

MEN AVAILABLE

479 Mechanical engineer or superintendent for large corporation; experience along broad mechanical and steam power lines; executive, with ability to handle large number of men, or will consider proposition to act as consulting engineer.

480 Mechanical engineer, 40 years of age, American, has had both practical shop and technical training; is expert in the up-to-date manufacture of interchangeable parts, resourceful in design and process for reducing the cost and increasing production, good organizer and system man with excellent executive ability, expert in the design and manufacture of commercial vehicles, both gasolene or electric, Edison or lead storage batteries. Can furnish best of references.

481 Member, possessing broad general experience in shop, drafting room, and executive positions, desires connection with large manufacturing company where ability in devising labor-saving appliances, machinery, and methods of manufacture will be of value. Part of remuneration preferably based upon percentage of savings effected in reduction of manufacturing cost.

482 Junior, M.I.T. graduate, three years experience in building, design and construction. At present employed as engineer by large industrial concrn. Desires position with firm of constructing engineers or building contractors.

483 Mechanical engineer, 33 years old, married, graduate of leading technical school, wishes position as manager of small growing plant. Three years experience as machinist apprentice and two years as draftsman, four years in industrial and production engineering, installing cost, shop and production systems rearranging and equiping plants for increased efficiency, standardizing products, operations and tools. Now employed as general superintendent of factory employing 150 to 200 men.

484 Mechanical and electrical engineer, ten years experience in the design, construction, operation and management of steam and water power plants for industrial and public service companies, desires permanent position with consulting or contracting engineers in or near New York.

485 Associate member, age 38 years, Lehigh University graduate, now in charge of the construction of a 9000-kw. lighting plant, will be open for employment in an executive capacity, as manager, engineer or superintendent, upon the completion of the present work, October or earlier. Sixteen years varied experience along mechanical, electrical and civil engineering lines, involving design, supervision, construction, operation, purchasing, etc., in connection with electric railways, power and industrial plants, etc. Good executive. Salary $3000.

486 Superintendent of power department or power plant. Wide experience covering twelve years in construction and operation. Expert on

modern steam turbine plants. Practical, systematic and economical. With present concern five years.

487 Member, with twelve years experience in jobbing and contract work, covering foundry, machine, boiler and plate, and structural steel. Experience from draftsman to superintendent. At present, sales engineer for large power house engineers and contractors.

488 Position as works engineer wanted by member, 34 years of age, of broad experience in power plant design, construction, operation, heating systems, building construction and general factory engineering.

489 Position desired by executive engineer, member, 36 years old, seventeen years broad experience in designing, building, erecting and operating, power generating, hydraulic, electric, crushing, grinding, conveying, power transmission and machine tool machinery; concrete and civil engineering. Past seven years executive positions in industrial plants, competent organizer and handler of labor, able to secure from both labor and machinery maximum economic output. Average salary for past four years, $5000. Present location, South West, excellent references from past connections.

490 Technical graduate, 30 years of age, Junior member, at present employed in locomotive shops, five years designing and estimating costs in the manufacture of railroad and contractors equipment. Four years with engineers and contractors on power and industrial plant design. Will consider position as assistant engineer salesman or branch manager. Personal interview can be arranged in New York or Philadelphia. Minimum salary at beginning, $1800.

491 Associate, aged 35, at present employed as checker in drawing room, desires position of some responsibility in Philadelphia or suburbs. Sixteen years experience in civil, structural and mechanical lines, mainly the last. Employed as draftsman, squad foreman, chief draftsman and checker at rolling steel and pipe mills, by-product coke ovens and on chemical apparatus, steel cars and locomotives.

492 Position as superintendent, assistant superintendent or mechanical engineer of manufacturing plant, twenty years experience covering coal-handling machinery, agricultural implements and house-heating apparatus. At present employed. Details and references furnished to those interested.

493 Mechanical engineer, technical graduate, desires position as assistant superintendent or designer. Can furnish good references.

494 Motor-truck engineer open for position as transportation expert and efficiency man; competent to investigate and report upon delivery system. Eight years' continuous experience in this industry in designing, constructing, testing motor trucks, and writing on truck subjects.

495 Position desired with mechanical trade paper by editor and writer of several years' experience. Sufficient previous practical training and technical education. Knowledge of cuts, printing, make-up and make-ready, all phases of publishing work. Competent to bear responsibility and direct others.

496 Technical graduate and member desires position as works manager or superintendent. Wide experience in manufacture of high-pressure boilers. At present employed.

ACCESSIONS TO THE LIBRARY

WITH COMMENTS BY THE LIBRARIAN

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary Am. Soc. M. E.

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———Standard Specifications Governing the Chemical and Physical Properties of Bessemer Steel Rails as adopted May 24, 1906.

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UNITED ENGINEERING SOCIETY

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EXCHANGE

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SPRAGUE ELECTRIC WORKS, *New York, N. Y.* Sprague electric grab-bucket cranes, 24 pp.

SAMU'L C. TATUNI & CO., *Cincinnati, O.* Drilling, punching, perforating of paper, 40 pp.

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Note—Numbers in parentheses indicate number of years the member has yet to serve.

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THE BALTIMORE SEWERAGE PUMP VALVE

By A. F. Nagle

ABSTRACT OF PAPER

This paper gives the results of experiments made by the Bethlehem Steel Company with a small experimental pump to determine the best proportions of the pump valve to be used in the Baltimore sewerage pumping engine. The experiments showed that a plain, flap valve with a seat angle of 60 deg. with the vertical plane, weighing about 0.26 lb. per sq. in. (in water) of seat opening, or 0.30 lb. in the atmosphere, measured normal to the face of the valve, will give a quiet and prompt working valve up to fully 30 r.p.m.

A discussion of the theory involved in the action of the valve, namely, that the velocity of the flow through the valve opening corresponded quite closely to the velocity due to this head or weight, held very fairly. Also that on opening, the valve lift was in perfect synchronism with the plunger velocity at any speed. In closing, the valve lagged behind for the greater part of its descent, but at the speeds given always reached its seat at the end of the stroke without shock.

THE BALTIMORE SEWERAGE PUMP VALVE

By A. F. Nagle, Bethlehem, Pa.

Member of the Society

In December 1906, the Baltimore Sewerage Commission awarded to the Bethlehem Steel Company a contract for building three sewage pumping engines of substantially the same design as the modern standard city water works pump, namely, "a self-contained, vertical, triple-expansion crank and fly wheel, three single-acting plungers, condensing engine; each pumping engine to have a capacity of 27,500,000 gal. in 24 hr., with an average total head of 72 ft. (7 ft. suction lift); and to have a capacity for 50 per cent greater speed for short periods." The only deviation from the water works design was in the pump valves. These, instead of being made of rubber discs, 3 in. or 4 in. in diameter, spring operated, were to be as follows:

2 "They shall be rectangular flap valves swinging on horizontal hinges, faced with rubber of the best quality for the purpose, and *weighted;* the valve seats shall be of hard, tough bronze, and *the planes of their faces shall be inclined.* (Italics are the author's.) The hinges of the valves shall be protected, by hooding or otherwise, from the direct flow of the sewage passing through the valves; suitable stops shall be provided to limit the opening of the valves at the extreme edges to 2½ in., and the clear waterway through either suction or discharge valve shall not be less than 145 per cent of the area of the plunger."

3 In the spring of 1907, when the drawings were well advanced, the author was appointed by the commission to be their mechanical engineer and inspector at the Bethlehem Steel Company's plant. It had been decided to make the normal speed of the engines of 5 ft. stroke and 20 r.p.m., which would be increased to 30 revolutions to comply with the contract capacity.

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This speed would necessitate a plunger 40¼ in. diameter to give the required volume of water. For the valve decks there had been tentatively planned thirty-four 4 in. x 12 in. and six 4 in. x 9 in. valves.

4 The sewage pump valve is far from being of a standard design. The commission's engineers had given the subject much study and made many inquiries before writing the paragraph

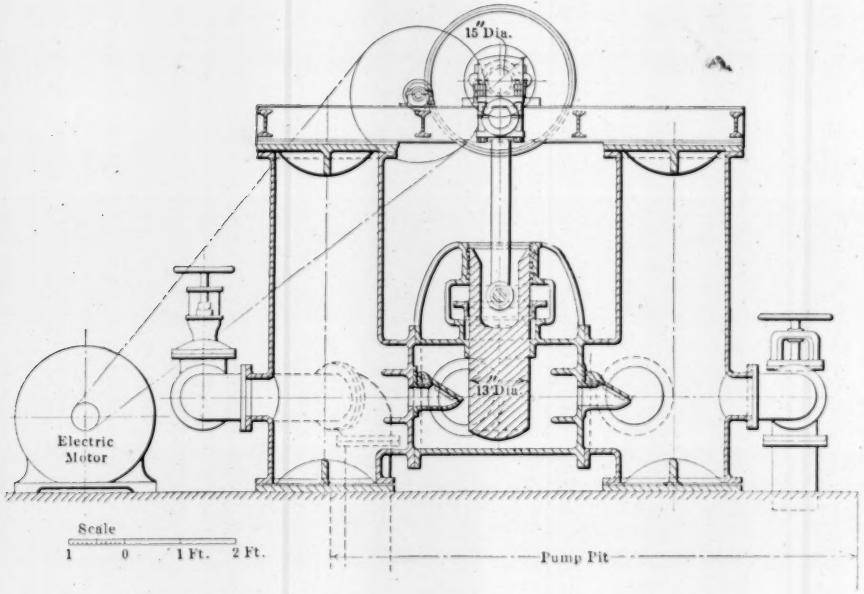

FIG. 1 PUMP FOR STUDYING VALVES DESIGNED FOR BALTIMORE SEWAGE

quoted, and yet it is noticeable that they refused to be specific on the essential elements of the valve, namely, *its seat-angle* and *its weight*. As there would be required over 700 of these bronze valves, it would be an expensive affair to make a change in them after completion. The writer therefore suggested to the management of the Bethlehem Steel Company that they construct a small pump for the purpose of trying out one set of the proposed valves. With this they were in hearty accord, and Fig. 1 shows a sectional elevation of the pump as made. A description is hardly necessary except to say that it is 13 in. in diameter by 15 in. stroke, making from 20 to 30 r.p.m.: being the exact equivalent of 6½ in. diameter by 5 ft. stroke, the proportional

ratio of plunger and valves. The 7 ft. suction lift was also reproduced.

5 In order to obtain definite knowledge of the action of the valve, there was attached to the arbor of its hinge a 5/8-in. shaft extending through the casing and a light stuffing box, and to this shaft were fastened suitable arms, levers and rods leading to an ordinary indicator from which the spring had been removed. In this way, a perfect trace of the lift of the valve could be obtained, examples of which accompany this paper. At the same time a pump card was taken and some of these are also reproduced.

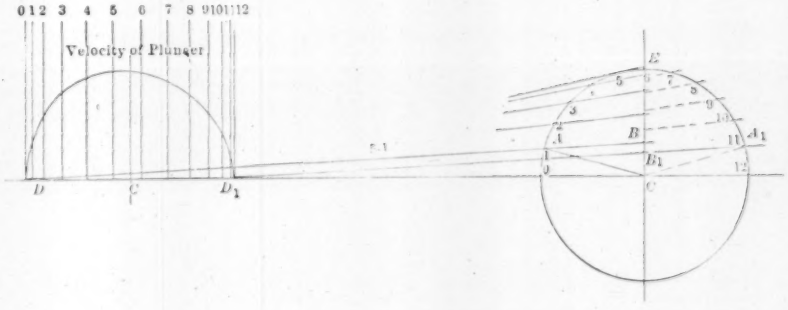

FIG. 2 GRAPHICAL METHOD FOR FINDING PLUNGER VELOCITY

6 The plunger velocity is dotted on the valve card, making the maximum plunger velocity agree with the highest lift of valve. The graphical method of finding this velocity is reproduced in Fig. 2, copied in substance from Rankine's Rules and Tables, page 236. The only difference in condition between this small pump and the main pump is that the relatively more rapid motion of the plunger in this case is in the lower end while in the main pump it will be at the upper end. The practical result of this arrangement is that the suction valve is favored in closing time, while in the main pump it will be the discharge valve thus favored.

7 It was decided to experiment with three different seat angles, 30, 45, and 60 deg. with the vertical plane, and weighted to different amounts. From the large number of tests made, two sets of cards for each of the three seat angles have been selected, at 20, 25, and 30 r. p. m.

8 The first test at a change of seat angle was always made with the normal or unweighted valve. Then followed one with

the suction valve loaded 5¼ lb., and the discharge valve 10.8 lb. There is no good reason, however, why the valves should not have been weighted alike.

9 Valve and pump cards are shown in Figs. 3, 4, 5, 6, 7, and 8, and data pertaining thereto are given in Table 1, Par. 23.

EXPLANATION OF TABLE 1

10 Columns 1 to 5 inclusive are self-explanatory. Columns

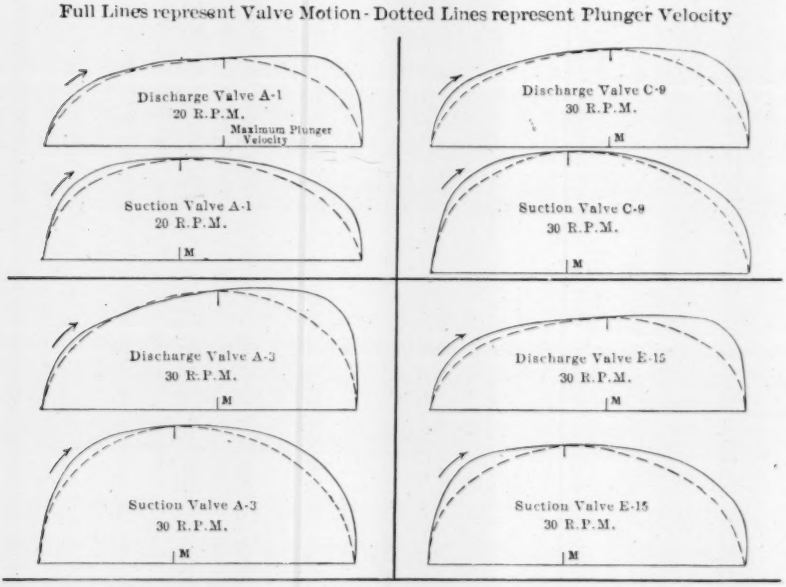

FIG. 3 VALVE CARDS, SEAT ANGLE 30 DEG.

6 and 7, weight of valve, is the entire weight before it is put in place. Columns 8, 9, and 10, 11, are obtained by a spring scale attached to the outer edge of the flap and noting the force required to start the valve; and again, the amount at its full lift of 2½ in., the scale being held, of course, at right angles with the flap. It should be noted that owing to friction of the hinge, it usually required from one to two pounds more to move the flap upward than downward, and the amount given in the table is the mean of the two.

11 Columns 12-13: The actual lift of the valve is obtained as follows: Owing to the intercession of certain cranks and levers,

the pump card trace is not exactly in line with the outer edge of the flap. To correct for this the valve cards as taken must be multiplied by the following factors: 1.25 for the 30 deg. seat; 1.25 for the 45 deg. seat; and 1.54 for the 60 deg. seat.

MECHANICAL ACTION OF THE VALVES

12 Mr. A. J. Mayer, on behalf of the Bethlehem Steel Company, took all cards and made the following comments on the working of the pump valves:

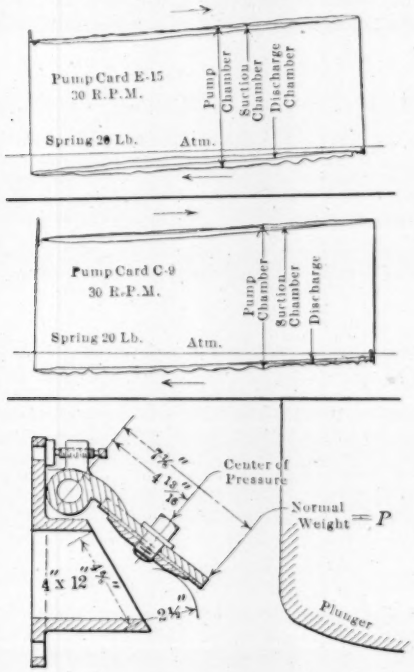

FIG. 4 PUMP CARDS, SEAT ANGLE 30 DEG.

Seat Angle, Deg.	Card No.	R.p.m.	Suction Valve	Discharge Valve
	A-1	20	Knocks slightly at closing. Very slight knock at opening.	Knocks at opening louder than suction. Very slight knock at closing.
30	A-2	25	Knocks slightly at closing. Very slight knock at opening.	Knocks at opening louder than suction. Very slight knock at closing.

Seat Angle, Deg.	Card No.	R.p.m.	Suction Valve	Discharge Valve
	A-3	30	Knocks slightly at closing. Very slight knock at opening.	Noises slightly increased at opening and closing.
	C-7	20	Barely audible knock at closing. No noise at opening.	Pounds at opening. No noise at closing.
30	C-8	25	Barely audible knock at closing. No noise at opening.	Pounds at opening. Barely audible pound at closing.
	C-9	30	Barely audible knock at closing. No noise at closing.	Noise slightly increased at opening. Very slight pound at closing.
	21	20	No noise at opening or closing.	Knock slightly at opening and closing.
45	22	25	No noise at opening or closing.	Noises slightly increased.
	23	30	No noise at opening or closing.	Knocks loud at opening and closing.
	27	20	No noise at opening or closing.	Barely audible pound at opening. No noise at closing.
45	28	25	No noise at opening or closing.	Knocks slightly at opening and closing.
	29	30	No noise at opening or closing.	Knocks slightly at opening and closing.
	39	20	No noise at opening or closing.	Barely audible thud at opening. No noise at closing.
60	40	25	No noise at opening or closing.	Audible thud at opening, and closing.
	41	30	No noise at opening or closing.	Pounds slightly at opening. Audible thud at closing.
	45	20	No noise at opening or closing.	Barely audible thud at opening. No noise at closing.
60	46	25	No noise at opening. Slight thud at closing.	Makes thud at opening and closing.
	47	30	No noise at opening. Slight thud at closing.	Makes thud at opening and closing.

Seat Angle, Deg.	VALVE DIMENSIONS	Square Inches
30	Area of Valve Seat Outside	82
	Area of Valve Seat Inside	55
	Ratio of Areas, 1 to 1.48	
	Area of Full 2½-in. Valve Lift	45
45	Area of Valve Seat Outside	101
	Area of Valve Seat Inside	72
	Ratio of Areas, 1 to 1.41	
	Area of Full 2½-in. Valve Lift	49.50
60	Area of Valve Seat Outside	107
	Area of Valve Seat Inside	80
	Ratio of Areas, 1 to 1.34	
	Area of Full 2½-in. Valve Lift	55

NOTE: It should be stated that owing to the nearness of the hub of the hinge to the body of the valve, only the front and two sides of the opening were taken as discharge area.

13 A point is now reached where there are facts to serve as a guide in the selection of a design for the valve, without any mathematical calculation whatever, the application of which is left to a later part of this paper. Referring for a moment to Mr. Mayer's remarks, attention should be called to the difficulty in distinguishing between a "thud" and a "knock" of a valve, either suction or discharge. The writer's opinion is that a "thud" always occurs at the *opening* of a valve, due to the difference between the inner and outer seat area, thus necessitating a greater pressure within to start to open the discharge valve than exists without; and the converse for the suction valve. All the pump cards show this greater or less pressure by a descent or an ascent of the indicator piston, which latter, however, falls to the normal pump pressure as soon as the valve opens. If there were air-pockets in a pump, the pressure of the air when released would send the valves violently against their stops. The soft rubber or leather facing allows this change of pressure to occur with nothing worse than a "thud."

14 In addition to the valve action, there is at this time the reversal of all stresses in the moving parts and journals, thus tending to increase the "thud" of the pump.

15 In contrast with a "*thud*" at *opening* is the "*knock*" at *closing*. This knock is due to the violent seating of the valve by the reversed action of the plunger, if, instead of coming gently

to its seat by its inherent weight as the plunger reaches the end of its stroke, it is not then seated.

SELECTION OF A DESIGN

16 A cursory examination of the valve traces or cards will show:

First, that *at opening*, every valve trace followed closely the theoretical or plunger-velocity line, except those

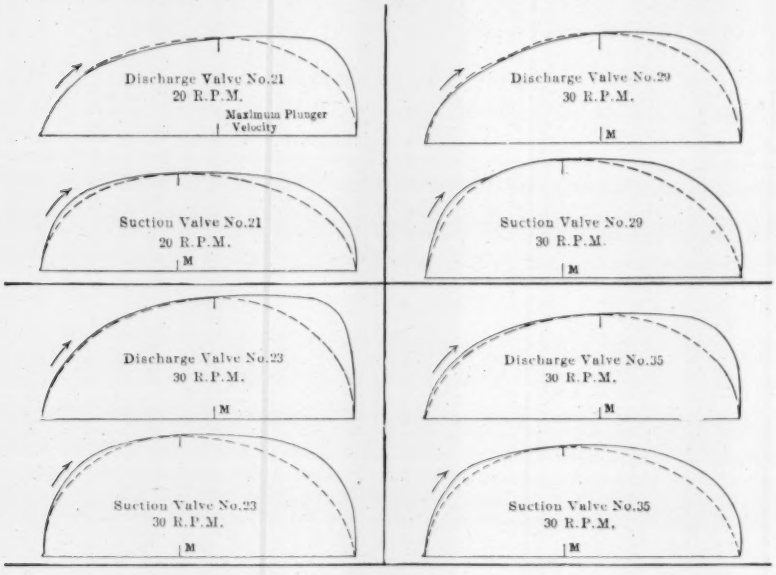

FIG. 5 VALVE CARDS, SEAT ANGLE 45 DEG.

of 30-deg. seat angle. In this case the valve acted like a swinging pendulum, in the impetus it received in opening, notwithstanding its natural increased weight, due to change of angle, carrying it beyond its proper position. In studying these cards, one must not lose sight of the different plunger velocities at the two ends of the stroke.

Second, in closing there was a constant improvement from the 30 deg. to 60 deg. seat angle; and, of course, with its increased weight. The heavier the valve the better it worked, mechanically, but the lift was reduced—

not desirable in a sewage valve—and the velocity through the valve increased.

17 In opening, every pump valve, either suction or discharge, if of constant or nearly constant weight, such as all gravity acting valves are, is *forced* by the plunger to follow its own varying velocities; but in *closing* there is no force or pressure to do this.

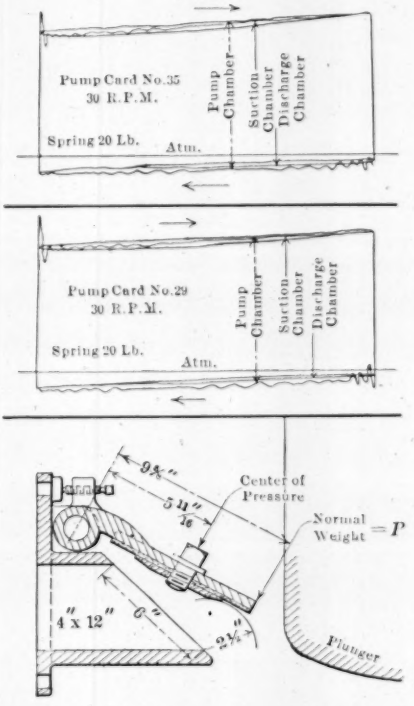

FIG. 6 PUMP CARDS, SEAT ANGLE 45 DEG.

It has only its own inherent weight to move it, and its effective weight is diminished by several causes. First, there is the resistance of its hinge, probably very small, since the fit is very free; second, the water cannot fill in the back of a flat valve any more than it can flow into the space left by the square stern of a boat; hence the partial vacuum formed there increases the lifting force of the stream beneath it; and third, there is the inertia of the mass of the valve as well as of the water which has to change its position.

18 Perhaps the experience all have had in rowing a boat up or down a rapidly moving stream will give a good impression of the difference between closing and opening a pump valve.

19 At 30 revolutions the 60 deg. seat valve (unweighted) reached only 2 in. of its 2½ in. possible lift. To have made it light enough to go to its 2½-in. stop at this speed would have necessitated reducing its weight about one-third (see last line in

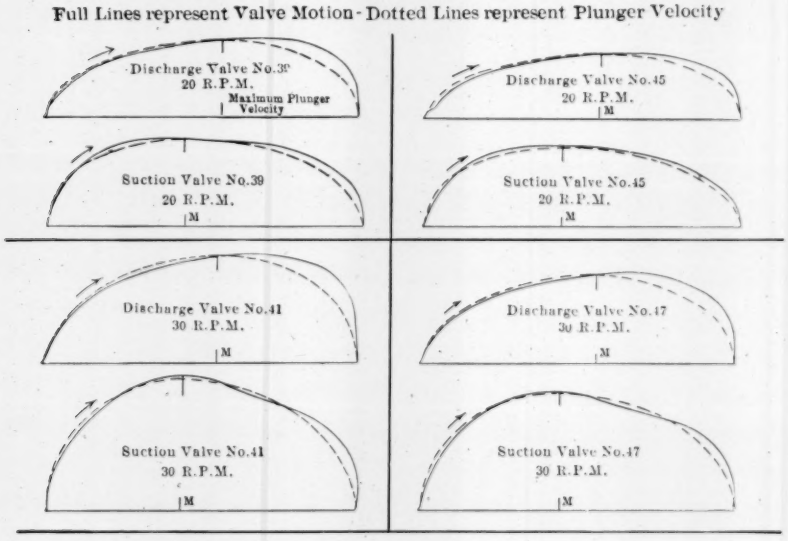

FIG. 7 VALVE CARDS, SEAT ANGLE 60 DEG.

Table 5) and with every indication that it would not reach its seat in closing in a way to prevent slamming. Valve cards 39-41 furnished the best proportions for a large lift and lowest velocity through the valve. Its weight of seat area per square inch was 0.221 lb. (See Table 3.) The extraordinary lift of 2½ in. of the suction valves 41 and 47 can be attributed to the proximity of the plunger at the lower end of its stroke, thus causing a back pressure at the valve outlet.

20 Fig. 9 is a reduced scale drawing of the valve as actually constructed. It weighs about 0.30 lb. per sq. in. of its seat area. It is of bronze throughout and has a renewable leather face. Its stop is cast on the body of the valve above it. There are 44 valves exactly alike, grouped into a 7-ft. steel valve deck. Figs. 11 and

12 are reduced photographs of these decks. Fig. 11 shows the valves closed and Fig. 12 the valves blocked open. Fig. 13 shows the pump end of the engine as it was erected in shop No. 2 of the Bethlehem Steel Company. The writer standing at the base, furnishes a scale to the pumps.

21 These pumps have now been in operation for several months and from the first turnover have worked so quietly one would hardly know there were any valves in them.

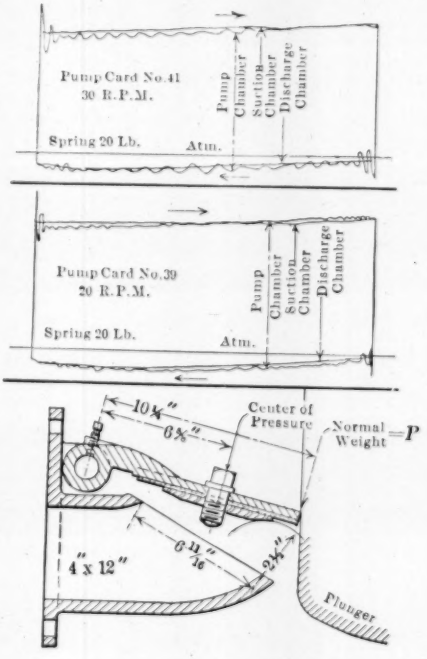

Fig. 8 Pump Cards, Seat Angle 60 Deg.

22 Aside from the design of the valve itself, great care was taken that all valves should be of like weight. When a group of valves are thus tied together, each is held in a very sensitive balanced position and any difference in weight is reflected in a different lift; hence a single light valve may be held back in closing and then slam to its seat. This was actually demonstrated by the action of the air-relief valve on top of the pump, which is a common swivel, swing check valve, without stop or definite weight.

23 Having described the valve as constructed, and conscious of its excellencies, shortcomings, cost of production and possible cost of renewals, the writer ventures to suggest that a cast-steel flap on a rounded seat would wear longer, cost much less to produce or repair and work fully as well, without a "thud," if the

TABLE 1 PUMP VALVE EXPERIMENTS. OBSERVED DATA

Seat Angle, Deg.	Card No.	R.P.M.	Pump Head, Ft.			Weight of Valve, Lb.		Normal Weight of Valve				Lift of Valve	
								Suction		Discharge			
			Suction	Discharge	Total	Suction	Discharge	Start	Top	Start	Top	Suction	Discharge
	1	2	3	4	5	6	7	8	9	10	11	12	13
30	*A*-1	20	7.59	67.1	74.7	27	28	8	11	8	10.5	1.60	1.38
30	*A*-2	25	7.71	65.8	73.5	27	28	8	11	8	10.5	1.85	1.63
30	*A*-3	30	7.93	66.9	74.8	27	28	8	11	8	10.5	2.15	1.90
30	*C*-7	20	7.25	66.9	74.2	32.2	38.8	11.5	14.5	14.0	17.5	1.46	1.14
30	*C*-8	25	7.71	65.6	73.3	32.2	38.8	11.5	14.5	14.0	17.5	1.65	1.29
30	*C*-9	30	7.93	65.0	72.9	32.2	38.8	11.5	14.5	14.0	17.5	1.92	1.53
45	21	20	6.35	65.9	72.2	30	31	10.1	11.1	10	11	1.50	1.50
45	22	25	6.35	66.7	73.0	30	31	10.1	11.1	10	11	1.70	1.69
45	23	30	6.35	64.6	70.9	30	31	10.1	11.1	10	11	1.96	1.94
45	27	20	7.03	67.1	74.2	35.2	41.8	13.7	15.2	16.7	19.5	1.36	1.24
45	28	25	7.03	67.9	74.9	35.2	41.8	13.7	15.2	16.7	19.5	1.50	1.38
45	29	30	7.03	66.8	73.8	35.2	41.8	13.7	15.2	16.7	19.5	1.80	1.64
60	39	20	6.80	67.3	74.1	34	32	13.2	14.2	12.7	13.6	1.69	1.48
60	40	25	6.80	66.1	72.9	34	32	13.2	14.2	12.7	13.6	2.20	1.81
60	41	30	6.80	66.1	72.9	34	32	13.2	14.2	12.7	13.6	2.50	2.00
60	45	20	6.80	64.5	71.3	39.2	42.8	17.6	19.1	21.1	22.4	1.54	1.25
60	46	25	6.80	65.6	72.4	39.2	42.8	.....	.....	.....	.....	1.92	1.51
60	47	30	7.36	66.1	73.4	39.2	42.8	.....	.....	.....	.....	2.26	1.68
			Aver.		73.3								

seat were of bronze or possibly of cast iron. Half a dozen of these valves, as shown in Fig. 10, have been made for future trial.

AGREEMENTS

24 Having described the essential facts leading to the selection of the design of this sewage pump valve, the writer does not like to leave the subject without an effort to present a theory as a guide to the construction of similar valves. Aside from the evidence given by the valve cards of the better working of the 60-deg. seat valve than of the valves with 30 deg. or 45 deg. angles,

there is the important result of a much larger outlet area with the same lift of valve, and consequent lower velocity through the valve. This is shown in the following tabulation of results with the unloaded valves at 30 revolutions:

30 deg. seats 34.2 sq. in. outlet, 1.90 lift, 7.72 ft. velocity
45 deg. seats 38.6 sq. in. outlet, 1.95 lift, 6.48 ft. velocity
60 deg. seats 44.4 sq. in. outlet, 2.00 lift, 5.94 ft. velocity

TABLE 2 PUMP VALVE EXPERIMENTS. CORRECTIONS OF ORIGINAL DATA APPLIED TO DISCHARGE VALVE. SEE FIGS. 14 AND 15

Seat Angle, Deg.	Card No.	R.P.M.	Lift of Valve		Normal Weight of Valve			
			Log, In.	Corrected In.	Log		Corrected	
					Start	Top	Start	Top
	1	2	3	4	5	6	7	8
30	*A*-1	20	1.38	1.43	8	10.5	8	10.5
30	*A*-2	25	1.63	1.61	8	10.5	8	10.5
30	*A*-3	30	1.90	1.90	8	10.5	8	10.5
30	*C*-7	20	1.14	1.14	14	17.5	14	17.5
30	*C*-8	25	1.29	1.29	14	17.5	14	17.5
30	*C*-9	30	1.53	1.54	14	17.5	14	17.5
45	21	20	1.50	1.46	10	11	10	11.7
45	22	25	1.69	1.68	10	11	10	11.7
45	23	30	1.94	1.95	10	11	10	11.7
45	27	20	1.24	1.20	16.7	19.5	16.7	19.5
45	28	25	1.38	1.38	16.7	19.5	16.7	19.5
45	29	30	1.64	1.61	16.7	19.5	16.7	19.5
60	39	20	1.48	1.51	12.7	13.6	12.7	13.6
60	40	25	1.81	1.75	12.7	13.6	12.7	13.6
60	41	30	2.00	2.00	12.7	13.6	12.7	13.6
60	45	20	1.25	1.26	21.1	22.4	21.1	22.4
60	46	25	1.51	1.47	21.1	22.4	21.1	22.4
60	47	30	1.68	1.68	21.1	22.4	21.1	22.4

In the latter case, the throat area of 48 sq. in. is very closely approached. (See Fig. 17.)

25 The basis of all hydraulic calculations is the feet head, or pounds pressure, per square inch of area, and consequent velocity of flow in feet per second. This relation is fixed absolutely by natural law and if theoretical results are not obtained, one must look for some diverting cause, such as contraction, change of direction, or friction. These last effects must be ascertained by experiment. For simple forms such as pipes, the values are

well established; but for irregular shapes such as these valve openings present, no data exist, and from the limited amount of information gathered during the experiments, the writer is unable to make a complete analysis of the valves. They do give *something*, however, as a guide in future construction, which the writer will endeavor to present.

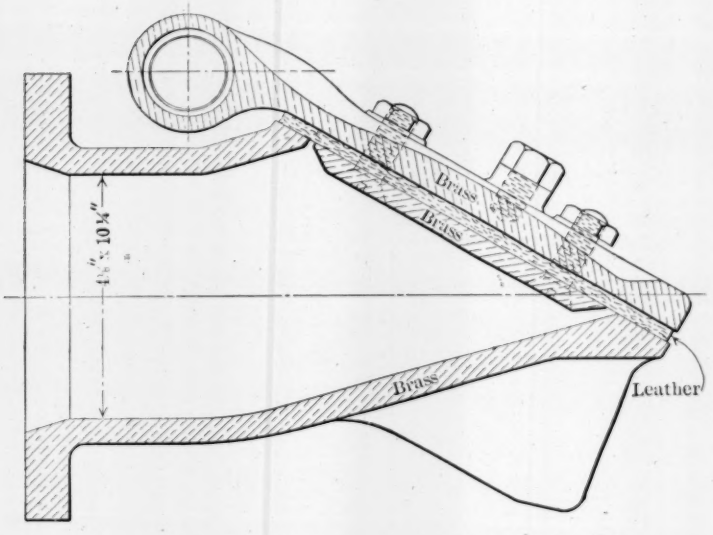

FIG. 9 VALVE AS CONSTRUCTED

26 The problems which it was attempted to solve are these:

First, knowing the weight—the normal weight—of a valve per square inch of seat area, how nearly will that weight, or pressure, give a velocity through the valve in agreement with the actual velocity found by experiment?

Second, knowing the lift of a valve at a given number of revolutions, how much will this lift be increased by an increase of revolutions?

Third, knowing the lift and weight of a valve at a given speed, what will be the lift if the weight be increased but the speed remain the same?

27 Before proceeding with the analysis it should be explained that it will be confined to the discharge valve only, and for these

reasons: In the first place, the writer has no theory of valve action that would distinguish between the two; second, certain discrepancies in the data, especially relating to the suction valve, led him to doubt their accuracy; and, third, there is no use in burdening the paper with more figures than are absolutely necessary.

28 As already explained, the original or log figures are given in Table 1. For the purpose of the calculations it has seemed

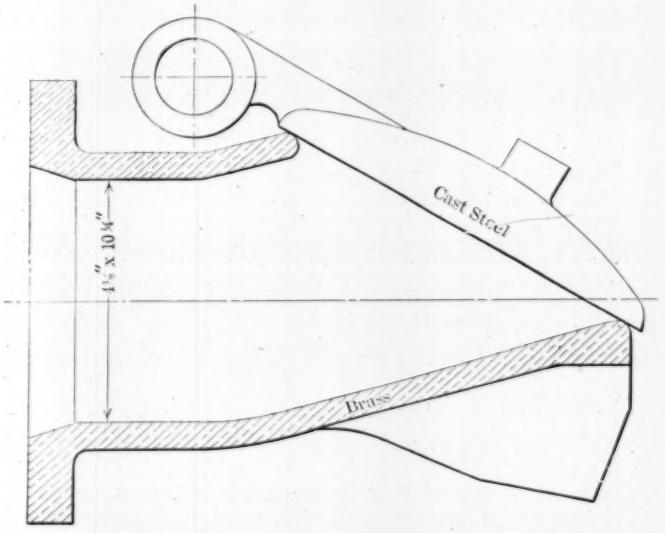

FIG. 10 DESIGN OF VALVE PROPOSED BY THE AUTHOR

necessary to make certain corrections upon these, as shown in Table 2 and in diagrams Figs. 14 and 15. Numerous small changes were made in the lift of the valve, but only one in the normal weight of the valve. These corrected values were used in all subsequent calculations, as exhibited in Table 3.

EXPLANATION OF TABLE 3

29 Column 14: The maximum plunger velocity is really slightly more than the crank velocity—for a 5 to 1 connecting rod it amounts to nearly 2 per cent, or 1.60 times the plunger travel, and this is given in this column. The main engine dimensions have been preserved rather than those of the experi-

mental pump, no change in calculated result being affected thereby.

30 Column 15: To facilitate subsequent calculations, there is given the plunger area corresponding to each valve (which is 33.00 sq. in.) multiplied by its velocity, giving the discharge per second in inch-feet.

31 Column 16: The area of valve opening is found by taking the full area opening given under "Valve Dimensions" in Par.

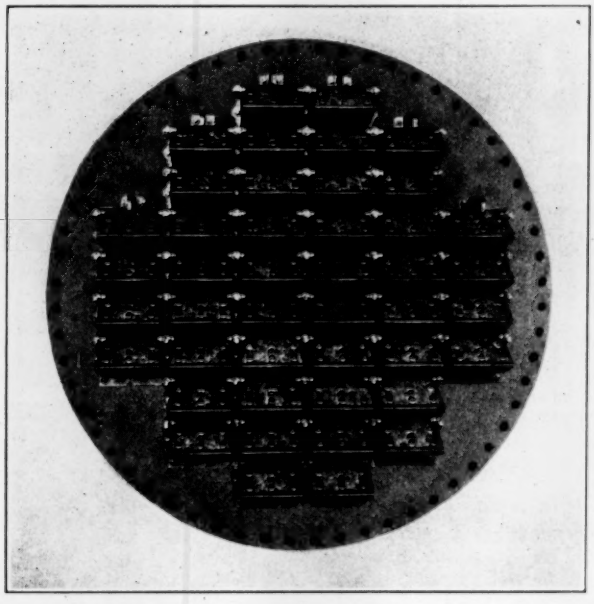

FIG. 11 VIEW OF VALVES, CLOSED

12, for 2½-in. lift, and reducing it to its proportionate amount for the actual lift given in Table 2, column 4.

32 Column 17: The apparent velocity through the valve is an important result. It is found by dividing the rate of discharge of plunger given in column 15, by the area of valve opening given in column 16. Therefore, if any two adjoining values in columns 16 and 17 be multiplied the corresponding result given in column 15 will be obtained. This is called the *apparent velocity*, although commonly called the *actual velocity*, for it is what the velocity would be if the area given were wholly filled with the issuing stream; but there must have been some con-

traction, and hence a correspondingly higher velocity. The writer is unable to divide this apparent velocity into the various measured causes that produce it, and with this explanation rests this part of the discussion because he is unable to carry the analysis any further with the data at hand.

33 Column 18: Hydraulic engineers always desire to express the relation of *head* to *velocity*. Hence, if in column 17 the velocity through the valve is given, what is its corresponding

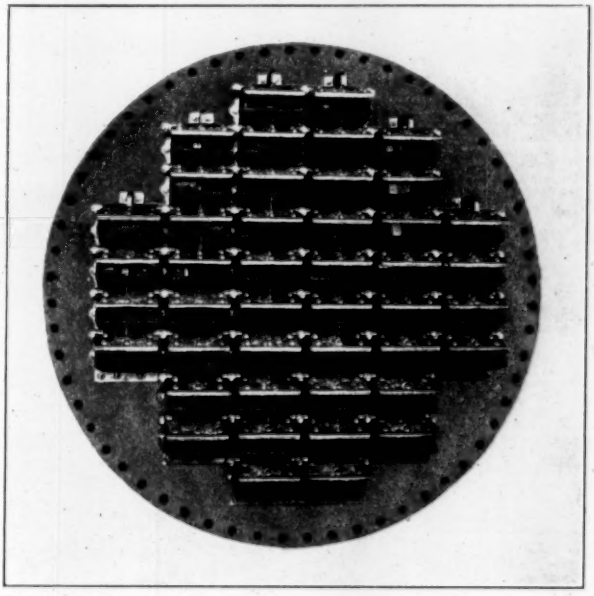

FIG. 12 VIEW OF VALVES, OPEN

head? Reducing the standard formula for feet head to pounds pressure, he has used the formula

$$\text{Vel.} = 12.20 \sqrt{P} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

P being the pressure in pounds per square inch.

34 Column 19: The normal weight of valve is given in Table 2, only at its two extreme positions. It is necessary to know what it is at the position of its lift. If, for example, it takes 8 lb. to lift valve A-1 at the start, and 10.5 lb. at its 2½-in. lift, the weight increases just one pound for each inch of lift. Hence, if the lift is 1.43 in. the weight at 1.43 in. lift will be 9.43 lb. These values are found in this column.

35 Column 20: The normal weight of valve per square inch of seat area is found as follows: In the previous column (19) is given the total normal weight at *the edge of the valve.* Taking the center of pressure of the issuing stream at the center of the

TABLE 3 PUMP VALVE EXPERIMENTS. CALCULATIONS BASED UPON OBSERVED DATA APPLIED TO DISCHARGE VALVE ONLY

Seat Angle, Deg.	Card No.	R.P.M.	Max. Plunger Velocity, Ft. per Sec.	Rate of Max. Pump Discharge, In-Ft. per Sec.	Area of Valve Opening, Sq. In.	Apparent Velocity through Valve, Ft. per Sec.	Velocity Head due to Velocity through Valve, Lb.	Normal Weight of Valve at Lift, Lb.	Normal Weight of Valve per Sq. In. of Seat Area, Lb.	Velocity due to Normal Weight of Valve, per Sq. In. of Seat Area, Lb.	Ratio of Velocities of Col. 21 to Col. 17
	1	2	14	15	16	17	18	19	20	21	22
30	*A*-1	20	5.33	176	25.7	6.85	0.315	9.43	0.247	6.08	1.13
30	*A*-2	25	6.66	220	29.0	7.59	0.387	9.61	0.252	6.12	1.24
30	*A*-3	30	8.00	264	34.2	7.72	0.400	9.90	0.260	6.22	1.24
30	*C*-7	20	5.33	176	20.5	8.58	0.494	15.6	0.409	7.81	1.11
30	*C*-8	25	6.66	220	23.2	9.48	0.603	15.8	0.415	7.87	1.21
30	*C*-9	30	8.00	264	27.7	9.53	0.610	16.2	0.425	7.95	1.20
45	21	20	5.33	176	28.9	6.09	0.249	11.0	0.222	5.75	1.06
45	22	25	6.66	220	33.7	6.53	0.286	11.2	0.224	5.77	1.13
45	23	30	8.00	264	38.6	6.48	0.282	11.3	0.229	5.84	1.11
45	27	20	5.33	176	23.8	7.39	0.366	18.0	0.363	7.35	1.01
45	28	25	6.66	220	27.0	8.15	0.436	18.2	0.367	7.39	1.10
45	29	30	8.00	264	31.9	8.28	0.460	18.5	0.373	7.45	1.11
60	39	20	5.33	176	33.2	5.30	0.189	13.2	0.219	5.71	0.93
60	40	25	6.66	220	38.5	5.71	0.219	13.3	0.221	5.74	1.00
60	41	30	8.00	264	44.4	5.94	0.237	13.4	0.223	5.76	1.03
60	45	20	5.33	176	27.7	6.35	0.270	21.8	0.363	7.35	0.86
60	46	25	6.66	220	32.3	6.81	0.311	21.9	0.364	7.36	0.90
60	47	30	8.00	264	37.0	7.14	0.342	22.0	0.366	7.38	0.94

seat opening (which is probably not exactly correct, but which is the best that can now be affirmed), it is needed but to reduce the extreme total weight to this center, divide by the seat area in square inches and correct for loss of weight in water.

36 For the 30 deg. seat are these values, P being the total normal weight at lift, found in column 19:

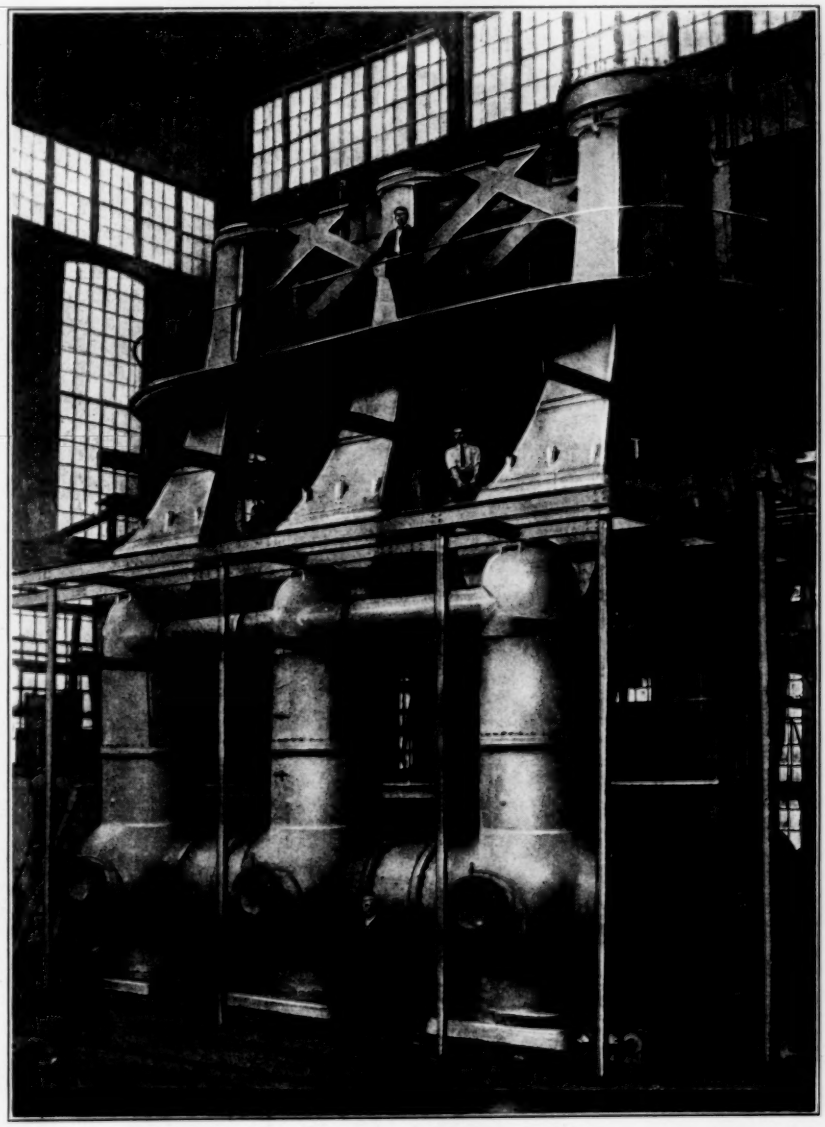

FIG. 13 PUMPS OF THE BALTIMORE SEWAGE PUMPING ENGINE AS ERECTED IN THE SHOP

$$W = \frac{7.87 \times 0.86 \times P}{4.81 \times 0.55} = 0.0256\ P \ldots\ldots\ldots\ldots [2]$$

For the 45 deg. seat these values become

$$W = \frac{9.62 \times 0.86 \times P}{5.69 \times 0.72} = 0.0202\ P \ldots\ldots\ldots\ldots [3]$$

For the 60 deg. seat

$$W = \frac{10.2 \times 0.86 \times P}{6.62 \times 0.80} = 0.0167\ P \ldots\ldots\ldots\ldots [4]$$

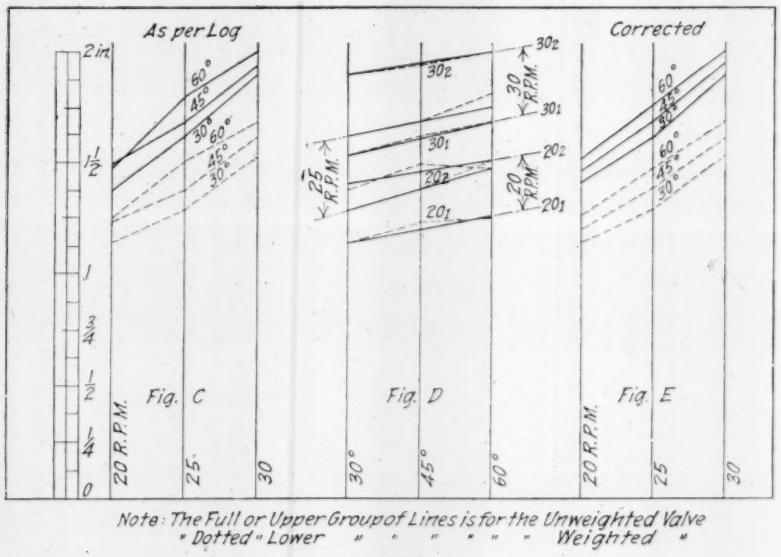

FIG. 14 LIFT OF DISCHARGE VALVE, CORRECTED. SEE TABLE 2

37 Column 21: *Velocity* due to weight of valve in water per square inch of seat area, as given in column 20, is found by formula [1].

38 Column 22: It now remains to see how closely these calculated velocities agree with the apparent velocities as given in column 17. These ratios are found in this column, and for better study they are plotted in Fig. 16.

39 A check on the ratios of velocities in column 22 can be had by finding a similar ratio of the weight of the valve and weight or "head" due to the velocity through the valve, as given in columns 18 and 20, bearing in mind, however, that the velocities vary with the square root of the pressure or "head." For

example—taking the first line in Table 3, columns 18-20, the weights are respectively as 0.315 to 0.247, and their square roots as 0.561 to 0.500, or 1.12 to 1, practically the same as found in column 22; these slight divisions being due to the number of decimals carried in the calculations.

COMPARISONS

40 The point has now been reached where the results of these experiments can be considered with some degree of intelligence.

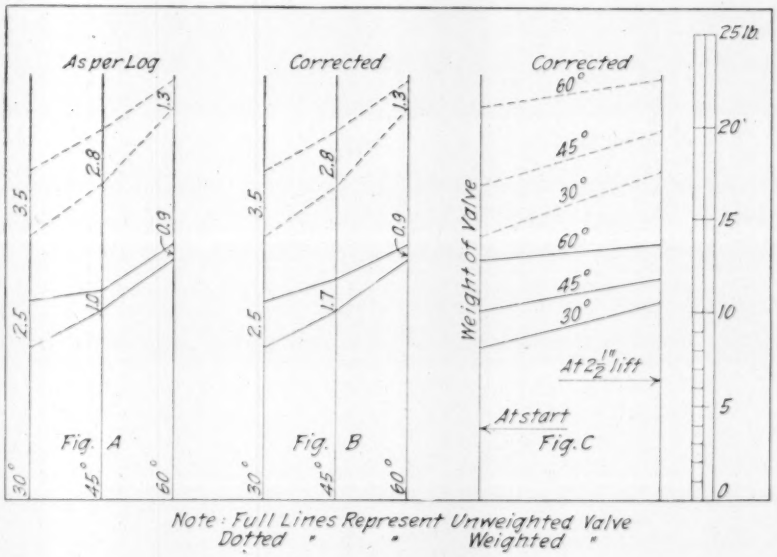

FIG. 15 TOTAL NORMAL WEIGHT OF VALVE AT START AND AT 2½ IN. LIFT. SEE TABLE 2

Turning to the first proposition propounded, How nearly does the calculated velocity, based upon the method here described, agree with the actual velocity through the valve as found by these experiments? On the whole, it seems to do so very closely: indeed, in the case of the 60 deg. seat and light valve, almost exactly. Rather than attempt to give mathematical expression to these deviations and apparent discrepancies, the designing engineer will use his own judgment as to the proper allowance to be made for them. Sufficiently accurate information has been obtained to prevent future mistakes in the construction of this type of valve as applied to the described conditions.

41 Second, In what proportion does the lift of the valve change with the speed? In Table 4 these results are tabulated and reduced to percentages, where they can be studied. The comparatively rough and uneven covering plate, holding the leather in place, is largely responsible for these erratic deviations. With

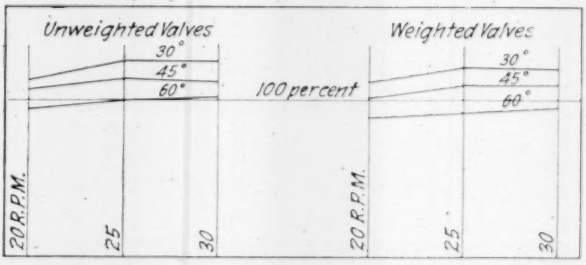

FIG. 16 RATIO OF AGREEMENT BETWEEN CALCULATED AND ACTUAL VELOCITIES THROUGH THE VALVE. SEE COL. 22, TABLE 3

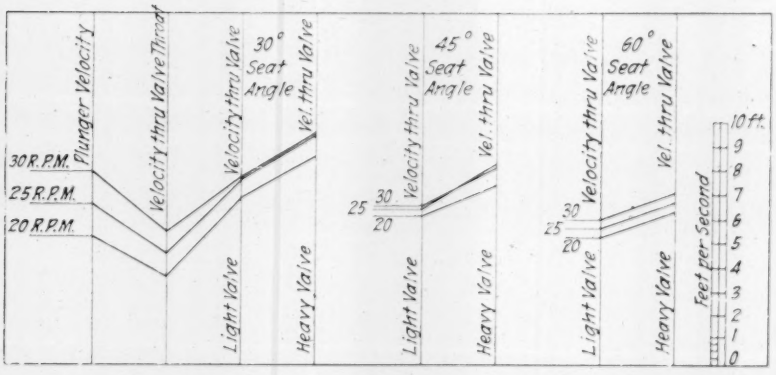

FIG. 17 MAXIMUM VELOCITY, FEET PER SECOND, OF PLUNGER AND OF FLOW THROUGH THROAT OF VALVE AND THROUGH DISCHARGE VALVE AT 20, 25, AND 30 R.P.M. AND WITH 30, 45 AND 60 DEG. ANGLE OF VALVE SEAT

a smooth valve face, as shown in Fig. 10, the deviations would probably have been gradual and proportionate to the change of angle by the lift, and a value might be given to them for different seat angles, but that will not be attempted here.

42 Third, The speed remaining the same, how does a change of weight affect the lift of the valve? Table 5 in its first six columns groups together previously given data, and in column 7 is applied the theoretical law that the lift should be inversely pro-

portional to the square root of the weight of the valve. That is, taking the lighter valve and its lift, what should be the reduced lift by loading it? In every case, it will be observed, the valves lifted *higher* than they should, according to this rule. On the 30 deg. seat, this variation was only about 3 per cent: on the 45 deg. seat, about 5 per cent: and on the 60 deg. seat, about 7.6 per

TABLE 4 COMPARISON OF LIFT OF VALVE WITH INCREASED SPEED

THEORETICALLY, THE LIFT SHOULD BE DIRECTLY PROPORTIONAL TO THE SPEED

Seat Angle, Deg.	R.P.M.	Light or Unweighted Valve				Heavy or Weighted Valve			
		Actual, In.	Theoretical, In.	Deviation		Actual, In.	Theoretical, In.	Deviation	
				Per Cent	Per Cent			Per Cent	Per Cent
1	2	3	4	5	6	7	8	9	10
30	20	1.43	1.43	100	...	1.14	1.14	100	...
30	25	1.61	1.79	90	100	1.29	1.42	91	100
30	30	1.90	2.14	88	98	1.54	1.71	90	100
45	20	1.46	1.46	100	...	1.20	1.20	100	...
45	25	1.68	1.82	92	100	1.38	1.50	92	100
45	30	1.95	2.19	89	96	1.61	1.80	89	97
60	20	1.51	1.51	100	...	1.26	1.26	100	...
60	25	1.75	1.89	92	100	1.47	1.57	94	100
60	30	2.00	2.26	88	95	1.68	1.89	88	95
	40	2.50				2.50	47 revolutions		

NOTE.—These last valves would have gone to their stops at 40 and 47 r.p.m. respectively.

cent. Why should it have lifted higher than calculated? The only explanation seems to be that the increased weight reduced the angle of impingement of the issuing stream against the face of the valve, and that this change of angle was proportionately more on the 60 deg. seat than on the others.

NOTES

43 In Fig. 18 are shown the areas of the plunger displacement and the valve areas. At once one may ask why these enlarged valve areas are greater than that of the valve opening, or outlet; or perhaps more pertinently, Why does not the engine

builder open the valve so as to give this required area when specifications call for, say, valve area of 145 per cent of plunger area? Simply because he cannot do it. If the valve were made light enough to open to this height, it would open, but it would close with a slam that would sound like a 10-in. gun explosion.

44 In Fig. 17 are plotted the plunger velocities and velocities through throat and outlet of valve, and it will be observed that

TABLE 5 COMPARISON OF WEIGHT AND LIFT OF VALVE

THE LIFT SHOULD BE INVERSELY PROPORTIONAL TO THE SQUARE ROOT OF ITS WEIGHT. DISCHARGE VALVE ONLY

Seat Angle, Deg.	R.P.M.	Weight per Sq. In.		Lift of Valve			
		Unweighted, Lb.	Weighted, Lb.	Unweighted, In.	Weighted, In.	According to Rule	Deviation, Per Cent
1	2	3	4	5	6	7	8
30	20	0.247	0.409	1.43	1.14	1.11	1.026
30	25	0.252	0.415	1.61	1.29	1.25	1.028
30	30	0.260	0.425	1.90	1.54	1.49	1.036
45	20	0.222	0.363	1.46	1.20	1.14	1.050
45	25	0.224	0.367	1.68	1.38	1.31	1.051
45	30	0.229	0.373	1.95	1.61	1.52	1.054
60	20	0.219	0.363	1.51	1.26	1.17	1.074
60	25	0.221	0.364	1.75	1.47	1.36	1.078
60	30	0.223	0.366	2.00	1.68	1.56	1.076
60	30	0.165	.....	2.50	....	....	.....

NOTE.—This last valve, weighing 0.165 lb. per sq. in. of seat area, making 30 r.p.m. would have gone to its 2½ in. stop.

with the lightest valve, on a 60-deg. seat, the velocity through the valve reached 5.30 ft. a second, while the plunger made 5.33 ft. a second. And yet, late water works specifications contain the clause that "the velocity through the valve shall not exceed 3 ft. a second!" Such a low velocity was never obtained through a pump valve. It would call for a valve weighing only 0.06 lb. per sq. in. of seat area—1/10 of the actual pressure or weights of valves, in practice. The writer has weighed numerous valve springs on modern city pumping engines and found them to be about 0.60 lb. (more and less) per sq. in. of net seat area at the

start, and tightening up to as much as 1.50 lb. when lifted to a stop only 9/16 in. high, entailing a velocity of at least 9 ft. to 15 ft. a second.

45 Turning to valve card No. 41, it will be seen that it closes promptly even at 30 r. p. m. with a 2 in. lift, and its weight at the start was only 0.212 lb. per sq. in. and 0.227 lb. at its full 2½ in. lift, an increase of weight of only 7.1 per cent. This is

TABLE 6 VALVE RESISTANCE BY INDICATOR CARDS AND BY CALCULATION. SEE FIG. 19

Seat Angle, Deg.	R.P.M.	Unweighted		Weighted	
		By Indicator Per Cent	By Weight of Valve Per Cent	By Indicator Per Cent	By Weight of Valve Per Cent
1	2	3	4	5	6
30	20	2.75	2.31	4.10	3.83
30	25	3.09	2.37	3.76	3.89
30	30	4.29	2.38	2.91	3.98
45	20	2.85	2.08	3.79	3.40
45	25	3.46	2.10	4.12	3.44
45	30	4.17	2.15	4.49	3.50
60	20	2.95	2.05	3.45	3.40
60	25	3.27	2.07	2.95	3.41
60	30	2.63	2.09	3.44	3.43
Averages.....		3.27	2.18	3.67	3.59

an initial weight of only about one-third and a final weight of only about one-sixth as much as is put into spring-operated pump valves.

VALVE RESISTANCE

46 It will be noted that the pressure line in both suction and discharge chambers was traced on the pump indicator cards. From these, the valve resistance commonly called friction could have been obtained if an average of many cards could have been had; but the few taken for each experiment were too few to give rational results. In Table 6, columns 3 and 5 are copied from Mr. Mayer's records and plotted in Fig. 19. The fluctuations are too great for any reasonable deductions.

47 The writer has therefore attempted to make a calculation of this resistance based upon the weight of the valve which, it seems to him, is the source of this loss. Practically, all other resistances within a pump are of relatively small amount, and

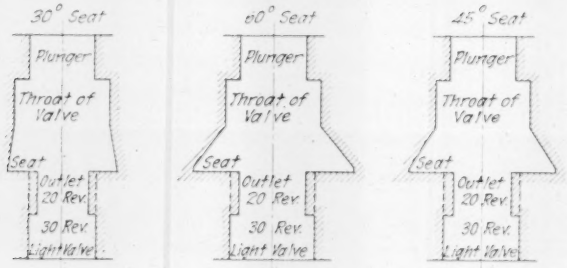

FIG. 18 PROPORTIONAL AREAS AT MAXIMUM VELOCITIES

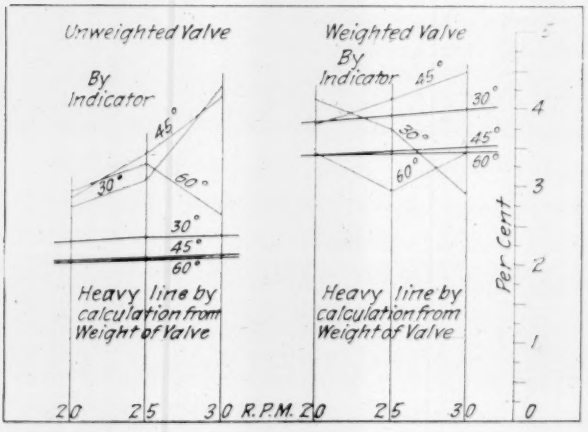

FIG. 19 PERCENTAGE OF VALVE FRICTION TO 72 FT. HEAD

hence the weight of a valve furnishes a reasonable criterion of pump losses.

48 It will be recalled that the apparent velocity through the valves is given in column 17, Table 3, and that no further allusion was made to the effect of contraction through the valve. If a calculation for loss of power is to be made, its amount will depend upon the amount of contraction, for the velocity must then be increased in an inverse ratio to the contraction. For the purpose of illustrating the method followed, a contraction of 81

per cent has been assumed. That would increase the velocity 23 per cent and the velocity head 50 per cent. Taking the weight of the valve as given in column 20 and increasing it by 50 per cent, gives a fair basis for calculating the loss of power within the pump. To bring this to a percentage basis it must be multiplied by two to apply it to both strokes and divided by 32 lb. for short (31.74 lb.) being the net total head in column 5, Table 1, and these values are given in columns 4 and 6, Table 6, and plotted in Fig. 19.

49 It must be left to the reader to decide whether the writer's assumptions and deductions are rational and justifiable.

50 The writer wishes to express his obligations to Mr. E. G. Grace, General Manager, Bethlehem Steel Company, for permission to make public the technical information obtained by these experiments.

THE VAUCLAIN DRILL

By A. C. Vauclain
and
H. V. Wille

ABSTRACT OF PAPER

In the usual type of drill there are four distinct cutting edges. The actual cutting, however, is done mainly by the two edges extending from the outer circumference to points near the center. At the central portion of the drill are the two shorter cutting edges constituting the "chisel point", which do not cut freely and introduce stresses tending to split the drill.

The paper describes a new type of twist drill which has two radial cutting edges and in which the usual chisel point of such drills is eliminated.

In this drill the stresses due to the chisel point are absent and the arrangement of the cutting edges is such that a greater thickness of metal is secured at the center of the drill, adapting it to heavy feeds by which means the most economical results are secured in drilling.

Tables are given showing the results of comparative tests with the Vauclain and regular types of drills, indicating a saving with the former of over half the horsepower expended with the latter type.

THE VAUCLAIN DRILL

By A. C. Vauclain, Philadelphia, Pa.

Non-Member[1]

and

Henry V. Wille, Philadelphia, Pa.

Member of the Society

Speaking generally, there can be no better definition of economical drilling than "rapid drilling"—the saving of time. The fact that a drill will cut at some phenomenal speed or will consume such and such an amount of power means nothing so far as productive capacity is concerned. The object in view is the removal of chips.

2 The productive possibilities of any metal-cutting tool are limited by its stress and heat resisting capacities. The temperature at which it will continue to operate successfully depends upon the excellence of material and manufacture, but the rapidity of heat generation and the stresses set up in the tool depend upon the design of the tool and the selection of feeds and speeds.

3 With a given excellence of material and manufacture, the strength of the tool must depend upon the extent and disposition of the section. The size of the drill is necessarily limited by the size of the hole to be drilled. The section of the drill has a lesser area than that of the hole, since space must be provided for the discharge of chips from the hole. The design of the drill should therefore be that which will give it the maximum of strength and strength conservation.

4 Cutting stress is practically independent of the cutting speed and with a given feed is proportional to the lip angle of

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The American Society of Mechanical Engineers, 29 West 39th Street New York. All papers are subject to revision.

the cutting edge. The cutting stress does not increase as rapidly as the feed.

5 The rapidity with which cutting heat is generated depends upon the cutting speed, the depth of feed and the lip angle of the cutting edge.

6 Since both the stress and heat are influenced by the keenness of the cutting-edge, it is desirable that the lip angle be as small as possible. But it must be blunt enough to carry off the

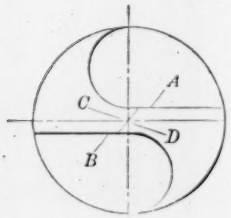

FIG. 1 COMMON TYPE OF TWIST DRILL

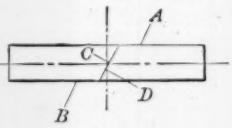

FIG. 2 COMMON TYPE OF FLAT DRILL

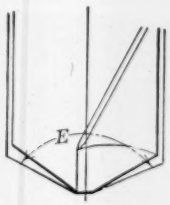

FIG. 3 SHOWING INCLUDED ANGLE OF CUTTING EDGES OF 118 DEG.

heat and to support the chip pressure, which falls more or less back of the actual cutting edge, according to the depth of feed. Since the chip is torn, not cut from the work, rupture between work and chip precedes the actual cutting edge. The heavier the feed, the farther back from the cutting edge will its pressure fall upon the tool.

7 Under ideal conditions, the torsional capacity of the drill should be its limit of cutting strength. This does not obtain in the present commonly used types of drill, and they break down very considerably below their torsional strength. Why this is so will be explained later in detail.

8 The feed remaining constant, the horsepower consumption

will be proportional to the speed. This is true both of the power consumed by the machine and that consumed in cutting. The speed remaining constant, the power consumed in cutting does not increase as rapidly as the feed and the power consumed by the machine remains constant for all feeds.

9 From this, it will be seen that the most economical method of chip production is by giving preference to the feeds, rather than the speeds. Power, time and drills will be saved thereby.

COMMONLY USED TYPES OF DRILLS

10 Figs. 1 and 2 illustrate the section scheme of drills now commonly used. While there are many modifications of these,

FIG. 4 ORDINARY TYPE OF DRILL SHOWING HOW DRILL SPLITS CENTRALLY IN HEAVY USAGE

the figures suffice to illustrate their common characteristic, which is that the cutting edges *A* and *B* pass to one side of the axis of motion of the drill instead of through the axis. In this respect there is no difference between Figs. 1 and 2.

11 It will be seen that in this scheme the drill has four distinct edges, *A*, *B*, *C* and *D*, and that the usual name given to it of "two-lip" drill is not correctly applied.

12 Referring to Fig. 3, it is customary for the included angle *E* to be of 118 deg. and the cutting edges *C* and *D*, Figs. 1 and 2, therefore have an unfavorable lip angle. These edges constitute what is commonly called the chisel point and their cutting resistance is very great. The cutting edges *A* and *B* cut more freely than the cutting edges *C* and *D* and a tendency to longitudinal fracture of the drill is set up thereby. This is the cause of the splitting of drills (Fig. 4). In explanation of this splitting tendency, Fig. 5 represents a section of brittle wire *A* held in vise jaws *B*. If sufficient pressure is applied, as indicated by arrows *X* and *Y*, the wire will break at *R* and *S*, due to the resistance

of the vise and to the forces X and Y tending to revolve it about its axis.

13 Those portions of the wire not within the vise jaws represent, in the drill, the cutting edges A and B, Fig. 1, free cutting, due to their favorable lip angle. That portion of the wire held within the jaws represents the cutting edges C and D, or the chisel point, imbedded in the work and having to overcome a high cutting resistance due to the unfavorable cutting lip angle of these edges. Under very heavy feeds, these two edges tend to stand still, while edges A and B continue to cut with the result that the drill is fractured in a manner similar to the breaking of the wire in Fig. 5. In the drill the width of the vise jaws becomes infinitesimal and the fractures R and S coincide and constitute the longitudinal splitting of the drill.

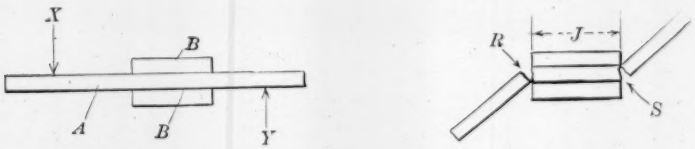

FIG. 5 ILLUSTRATING SPLITTING ACTIONS BY MEANS OF WIRE HELD IN VISE

14 In the commonly used types of drills there are but two methods of reducing the chisel point, viz., by thinning the drill at its center, or by pointing. By the former method the resistance to longitudinal splitting becomes lowered and by the latter method the cutting edges lose their support at and near the center of the drill.

15 By the foregoing it will be seen that the feed possibilities of the ordinary types of drills are not very great. Either the drill will split or its cutting edges will break under heavy feeds, long before its torsional capacity is reached. Summed up, the disadvantages of this type of drill are as follows:

a Its weakness.

b The unfavorable cutting lip angles of the chisel point create a tendency to longitudinal splitting of the drills.

c If the chisel point is reduced by central thinning of the drill, the resistance to longitudinal splitting is lowered.

d If the chisel point is reduced by pointing, the cutting

edges are deprived of their essential support at and near the center of the drill.

e Essential central thickness and cutting edge support are obtained only in connection with a considerable chisel point.

f The tendency to longitudinal splitting is increased by any increase in the extent of the chisel point.

g The resistance to penetration due to the chisel point is very great and is increased by any increase in the extent of the chisel point.

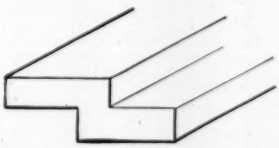

FIG. 6 DIAGRAM ILLUSTRATING PRINCIPLE OF VAUCLAIN DRILL

FIG. 7 CENTRAL EDGES OF FIG. 6 SHOWN BEVELED

h It is not adapted to heavy feeds, and must therefore be used under the conditions of moderate feeds and high speeds, the least economical method of chip production.

i There are four cutting edges, two of which do not pass through the axis of motion, hence the aggregate length of the cutting edges is excessive. This is productive of greater torque, hence of power consumption.

DESCRIPTION OF THE VAUCLAIN DRILL

16 The Vauclain drill is a "heavy feed" drill adapted to the most economical method of metal cutting and of great strength, due to its design.

17 In order better to describe the drill, the section scheme will first be explained diagrammatically and the development of the actual section will then be illustrated.

18 In Fig. 6 is shown a bar comprising two flat bars overlapping and integrally connected, as indicated.

191 By beveling, as shown at *O* and *P*, Fig. 7, the edges *A* and *B* are made to meet at the axis of the bar. Since these

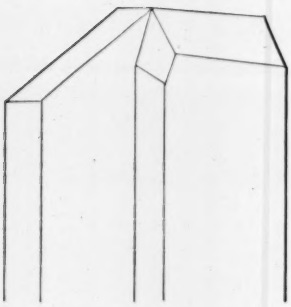

FIG. 8 SHOWING FIG. 7 GROUND TO FORM A DRILL

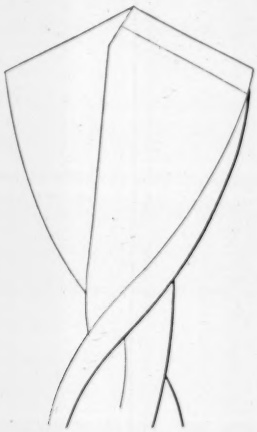

FIG. 9 SHOWING APPEARANCE OF DRILL WHEN TWISTED

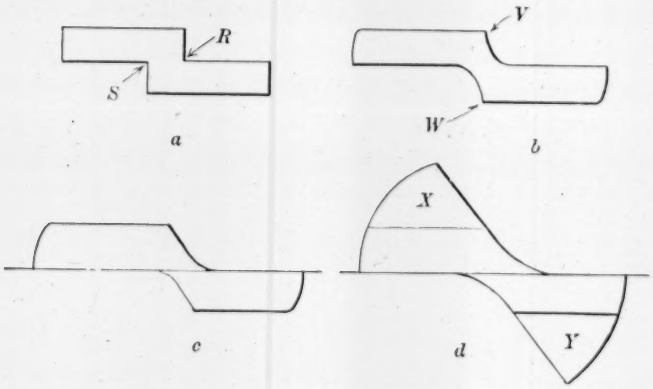

FIG. 10 SUCCESSIVE SECTIONS SHOWING DEVELOPMENT OF BAR OF FIG. 6 INTO FORM ADAPTED FOR VAUCLAIN DRILL

beveled surfaces are at an angle with the axis the integral connection between the two flat bars remains unbroken.

20 Fig. 8 indicates how the end of the bar would appear when ground to form a drill and Fig. 9 how it would appear if twisted.

21 Fig. 10-*a* again shows the section or end view of the bar in its original form. Introducing fillets at *R* and *S* gives the form shown in Fig. 10-*b* and removing the corners *W* and *V* modifies it as in Fig. 10-*c*. In Fig. 10-*d*, by adding areas *X* and *Y* to the section, its torsional value is increased.

22 Fig. 11 shows the exact section profile and in Fig. 12 is reproduced a photograph of three of these drills.

23 Tests have been made for the purpose of comparing the

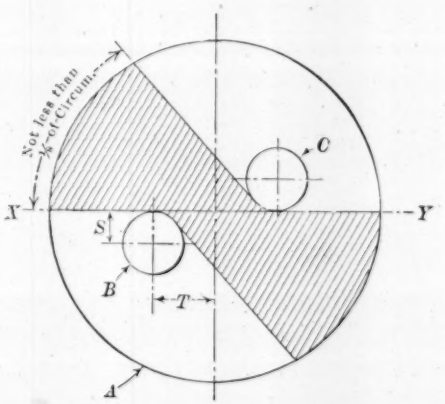

FIG. 11 DIAGRAM SHOWING EXACT FORM OF CONSTRUCTION OF DRILL

Circle *A* = Circumference of Drill
Diameter of Circle *B* = 3/16 Diameter of Drill
Diameter of Circle *C* = 3/16 Diameter of Drill
S = 3/32 of Diameter of Drill
T = 3/16 of Diameter of Drill
XY = indicates cutting edges

ordinary and the Vauclain types of drills. In order to secure uniform conditions care was taken that the different drill tests should be alike in quality of drill steel, heat treatment and tempering. The material drilled was a tough forging grade steel of about 0.45 carbon. Tables 1, 2 and 3 show the horsepower saved by giving the preference to the feeds, also the relative strength and endurance of the ordinary and the Vauclain types of drills. The size of drill used in all tests was $1^{9}/_{64}$ inches.

24 By examination of these diagrams and photographs it will be seen that the chisel point is eliminated without central thinning of the drill and without weakening the cutting edges.

ADVANTAGES OF THE VAUCLAIN DRILL

25 The improved design of this drill has brought about the following important features:

a Essential central thickness is secured.

b Cutting edges are properly supported at all points.

c Chisel point, with its unfavorable lip angles, is eliminated without weakening drill or cutting edges.

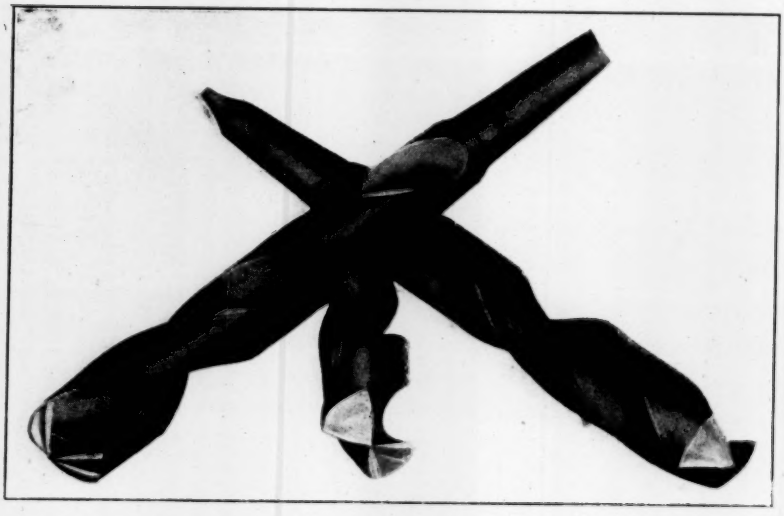

FIG. 12 EXAMPLES OF THE VAUCLAIN DRILL

d Cutting edges pass through axis of motion.

e There are only two cutting edges.

f Aggregate length of cutting edges is reduced to minimum.

26 As a result of these features the following advantages are secured:

g Increased strength and reduced liability to splitting.

h It is adapted to heavy feeds, the most economical method of cutting metal.

i Productive capacity is increased.

j Horsepower consumption per unit of metal removed is reduced.

k Life of drill is increased and frequency of grinding is less.

TABLE 1 STRENGTH COMPARISONS

Test	Kind of Drill	Feed per Rev.	Speed R.p.m.	Inches per Min.	Holes Drilled	Total Inches Drilled	Compare to Test	Condition of Drill After Test	See Note
1	Regular	0.009	300	2.70	1	3.875	6	O. K.	
2	Regular	0.0145	300	4.35	1	3.875	7	O. K.	
3	Regular	0.020	300	6.00	1	3.875	8	O. K.	*A*
4	Regular	0.030	300	9.00	1	3.875	9	O. K.	
5	Regular	0.050	300	15.00	0	0. —	10	Broke—see note	
6	Vauclain	0.009	300	2.70	1	3.875	1	O. K.	
7	Vauclain	0.0145	300	4.35	1	3.875	2	O. K.	
8	Vauclain	0.020	300	6.00	1	3.875	3	O. K.	*B*
9	Vauclain	0.030	300	9.00	1	3.875	4	O. K.	
10	Vauclain	0.050	300	15.00	1	3.875	5	O. K.—see note	
11	Regular	0.009	170	1.53	1	3.875	16	O. K.	
12	Regular	0.0145	170	2.465	1	3.875	17	O. K.	
13	Regular	0.020	170	3.40	1	3.875	18	O. K.	*C*
14	Regular	0.030	170	5.10	1	3.875	19	O. K.	
15	Regular	0.050	170	8.50	see note	1.000	20 note	Broke—see note	
16	Vauclain	0.009	170	1.53	1	3.875	11	O. K.	
17	Vauclain	0.145	170	2.465	1	3.875	12	O. K.	
18	Vauclain	0.020	170	3.40	1	3.875	13	O. K.	*D*
19	Vauclain	0.030	170	5.10	1	3.875	14	O. K.	
20	Vauclain	0.050	170	8.50	1	3.875	15	O. K.—see note	
21	Regular	0.009	170	1.53	1	3.875	26	O. K.	
22	Regular	0.145	170	2.465	1	3.875	27	O. K.	
23	Regular	0.020	170	3.40	1	3.875	28	O. K.	*E*
24	Regular	0.030	170	5.10	..	3.875	29	Broke emerging	
25	Regular	0.050	170	8.50	..	.....	30-31-32	.....	
26	Vauclain	0.009	170	1.53	1		21	O. K.	
27	Vauclain	0.0145	170	2.465	1		22	O. K.	
28	Vauclain	0.020	170	3.40	1		23	O. K.	
29	Vauclain	0.030	170	5.10	1	see	24	O. K.	*F*
30	Vauclain	0.050	170	8.50	1	note	25	O. K.	
31	Vauclain	0.050	208	10.40	1		25	O. K.	
32	Vauclain	0.050	245	12.25	see note	1.000	25	Broke—see note	

A. Drill broke after point had penetrated to 7/8 in. diameter, broke into many pieces; split. Torque is practically proportional to the square of the diameter, hence, since *A* broke when point had penetrated to only 7/8 in. diameter, *B* showed 170 per cent the strength of *A*.

B. Drill was permitted to emerge from work without knocking out feed; a slight crack in one cutting edge developed after drilling at rate of 15 in. per min. and emerging.

C. Drill broke into many pieces after penetrating 1 in. in last hole. Drilling at rate of 8.50 in. per min. split.

D. Since no heavier feeds were obtainable in the machine this drill was afterward run at 208 r.p.m. and 0.050 feed and broke only when emerging from bottom of hole; drilling at rate of 10.40 in. per min. and emerging. Taking into consideration that *D* emerged and *C* did not, the difference in strength is much greater than that indicated by the difference in inches per minute.

F. In order to get an actual comparison with *C* which broke at 1 inch penetration, this drill was not allowed to emerge from the work. It broke drilling at rate of 12.25 in. per min.; after penetrating 1 in. Drill broke torsionally and did not split.

l Distress in machine is less and resistance to penetration is reduced.

m Better holes are produced and afterwork of reaming is less.

TABLE 2 ENDURANCE TESTS

Test	Kind of Drill	Feed per Rev.	Speed R.p.m.	Inches per Min.	Holes Drilled	Total Inches Drilled	Compare to Test	Condition of Drill After Test	See Note
33	Regular	0.009	300	2.70	42	162.75	34	Cutting edges worn	A
34	Vauclain	0.020	208	4.16	42	162.75	33	Cutting edges worn	A
35	Regular	0.020	300	6.00	21	81.37	36	Cutting edges burned	B
36	Vauclain	0.030	208	6.24	21	81.37	35	Cutting edges O.K.	
37	Regular	0.020	300	6.00	6	23.25	38–39	Badly burned	
38	Vauclain	0.030	208	6.24	26	105.75	37	Cutting edges O.K.	C
39	Vauclain	0.030	208	6.24	35	135.12	37	See note	D
40	Vauclain	0.0145	208	3.02	20	...	41	O. K.	
41	Regular	0.009	208	1.87	20	...	40	O. K.	
42	Regular	0.009	300	2.70	11	...	43	O. K.	
43	Vauclain	0.0145	245	3.55	11	...	42	O. K.	
44	Regular	0.009	300	2.70	10	...	45	O. K.	
45	Vauclain	0.0145	245	3.55	10	...	44	O. K.	
46	Vauclain	0.020	245	4.90	5	...	47	O. K.	
47	Regular	0.0145	245	3.55	5	...	46	O. K.	
48	Regular	0.0145	300	4.35	10	...	49–52	O. K.	
49	Vauclain	0.020	300	6.00	10	...	48	O. K.	
50	Regular	0.020	300	6.00	25	...	51	Burned corners	B
51	Vauclain	0.020	300	6.00	39	...	50	Burned corners	E
52	Vauclain	0.020	300	6.00	10	...	48	O. K.	

A Cutting edges of Vauclain drill *less* worn than regular drill.

B Chisel point also burned.

C Broke in 26th hole by running into another hole.

D Spindle of machine ran hot, causing "jumping" of speed, and cutting edges; broken by resulting shocks. Drill at one period stood still, but did not fracture except at cutting edges.

E Point O. K.

TABLE 3 COMPARATIVE TESTS OF REGULAR AND VAUCLAIN DRILLS

Test	Kind	Feed per Rev.	Speed, R.P.M.	In. per Min.	H.P. Expended	Per Cent Saving
53	Regular	0.00599	300	1.797	10.86	....
54	Regular	0.02	87	1.74	10.30	....
55	Vauclain	0.02	87	1.74	5.03	51.2
56	Vauclain	0.00599	300	1.797	4.73	56.4

TESTS OF A 1000-H.P. TUBES HIGH B. & W. BOILER

By B. N. Bump

ABSTRACT OF PAPER

The tests described in this paper were made on an inclined header Babcock and Wilcox boiler, with superheater and mechanical stoker. The experimental unit was installed for the purpose of determining the maximum heat efficiency and fuel economy, and the variation of such efficiency with changes in the boiler rating, also to gather data useful in the design of a unit for a large installation. The peculiar feature of the boiler is the unusual height; the sections being 24 tubes high.

A Foster superheater was placed between the top row of tubes and the boiler drums, and the gases entered the superheater after passing over 24 tubes. The superheat was found to go up with the boiler rating, increasing from 30 deg. to about 75 deg. fahr. for a change in rating from 55 to 120 per cent.

The fuel was burned on a six retort Taylor stoker having a grate area, exclusive of dump grate, of 62.5 sq. ft. One of the results of increasing the height of a boiler is a large ratio of boiler heating surface to grate area, which in this case was 160 to 1. Such a large ratio limits the capacity of the boiler, making it impossible to run at high boiler ratings.

The best combined efficiency, 81.3 per cent, was obtained at about 56 per cent of rating. The efficiency decreases slowly with an increase in the boiler rating. The low rating tests show exit gas temperatures but little above the temperature of the saturated steam in the boiler. The radiation losses are low considering the exposed condition and the low atmospheric temperatures surrounding the unit.

TESTS OF A 1000-H.P. 24 TUBES HIGH B. & W. BOILER

By B. N. Bump, Syracuse, N. Y.,

Member of the Society

The boiler tested was erected as an experiment for the purpose of determining, by test and by continuous operation in the regular service of a large plant, the advantages and disadvantages of so high a boiler of the B. & W. type. These tests were made from November 1911 to February 1912. They were expected to answer the following questions:

a At what rating will the highest efficiency be obtained

b Will the superheat obtained, with the superheater located above the 24th tube, be sufficient to pay for the installation of the superheater

c Will the last pass of the boiler be effective

d Can the exit gas temperature be reduced very nearly to the temperature of the steam in the boiler

e If the exit gas temperatures are close to the steam temperature, will pitting occur in the back end of the boiler due to the sulphur content of the gas

BOILER AND SUPERHEATER

2 The boiler was built up of one regular 14-tube section and above this, with a space of some 11 in. between them, was placed a 10-tube section. The headers of the 10-tube and 14-tube sections were joined by short nipples, making a 24 high header. There are 21 of these sections containing a total of 504 tubes. The tubes are 4 in. in diameter, and 18 ft. long. There are 3 drums 42 in. in diameter, and 24 ft. long. The water heating surface of the boiler is 10,000 sq. ft. Between the top tubes and

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the drums a Foster superheater is placed containing 1750 sq. ft. of superheating surface.

3 The gases make three passes through the boiler. The baffles are arranged to give a gradual decrease in space through the first and second passes in the direction of flow, which tends to maintain the velocity of the gases as the temperature decreases. The gases enter the superheater from the top of the first pass, after traveling over 24 tubes. From the superheater they turn downward through the second pass; upward through the third pass, and then make their exit between the downtakes, and through the rear wall to the stack. The spaces between the header, both front and rear, were packed with asbestos.

STOKER AND FURNACE

4 The boiler is fired by a six-retort Taylor stoker, grate area 62.465 sq. ft., not including any dump grate area. The ratio of boiler heating surface to grate area is 160 to 1. The ratio of boiler plus superheater surface to grate area is 188 to 1.

5 The height of the combustion chamber is about 6.5 ft., and the width of the furnace 12 ft. 7 in., which latter is greater than the width of the stoker to allow for placing a cast-iron wind box on either side of the stoker. This is believed to be a new feature, tried as an experiment. Previously there had been trouble from the formation of clinker on the side walls with stokers of both the underfeed and overfeed type, owing to some coals giving a tough sticky clinker which is very troublesome. With the side walls against the stoker, clinker forms rapidly, and is difficult to remove. Setting back the side walls lessens clinker formation to some extent, and facilitates the removal of the clinker.

6 A brick ledge between the stoker and side wall is objectionable, because in time clinker will fasten to it and it is almost impossible to get a bar between clinker and ledge. The cast-iron wind boxes overcome the objections to the brick ledge by chilling the clinker so that it does not stick. A bar can always be forced between the clinker and wind box to raise the clinker. A part or all of the air from the blast fan is passed through these wind boxes on its way into the stoker. After six months of continuous service there is no sign of burning on the wind boxes.

7 Fig. 1 shows a front and side elevation of the unit. On the side elevation the locations of the pyrometer rods and gas sampling tubes are shown. The furnace temperatures were taken

through the three openings shown just under the lower tubes. The other gas temperatures were taken at positions 1 to 7 inclusive. The numbers on the data sheets and charts of gas temperatures refer to these positions.

8 The walls were lined with 9 in. of fire brick backed up by 9 in. of porous insulating brick. The outer surfaces of all walls were painted with two coats of boiler pitch to prevent any filtration of air through the brick work. Against these painted surfaces of the front and side walls 85 per cent magnesia covering 4 in. thick was placed. The magnesia was held against the brick and protected by a thin steel shell. Asbestos board was placed on the top half of the drums, then loose mineral wool was placed over the entire top of the boiler.

TEST APPARATUS AND DATA

9 The tests varied from 18 to 48 hours in length, although most of them were about 24 hours long. The coal was weighed in a wheelbarrow on scales tested frequently for accuracy. A small sample of coal was taken from each wheelbarrow load to make up the test sample for moisture, analysis, and calorific value.

10 The feedwater was measured in a tank of 13,000 gal. capacity. All feedwater piping was under observation, and any leaks were quickly discovered. The drips from the pump plungers were caught and returned to the feed tank. The feedwater piping being entirely independent there was no possibility of the measured water going anywhere except into the boiler. The blow-off piping was slip blanked, and the boiler blown down only between tests. The boiler was examined for leaks at frequent intervals, and the boiler tubes were dusted every 48 hours in the intervals between tests.

11 Steam pressures, superheated steam temperatures, and steam calorimeter temperatures were taken by recording instruments.

12 It was found necessary to have some suction over the fire to prevent the escape of gas into the boiler room. This suction was kept as low as possible. Gas temperatures were taken at seven different points as given in Table 5. The seven pyrometer rods were wired to one indicating instrument so that, by the use of knife edge switches, the indication of one rod after another could be taken in quick succession.

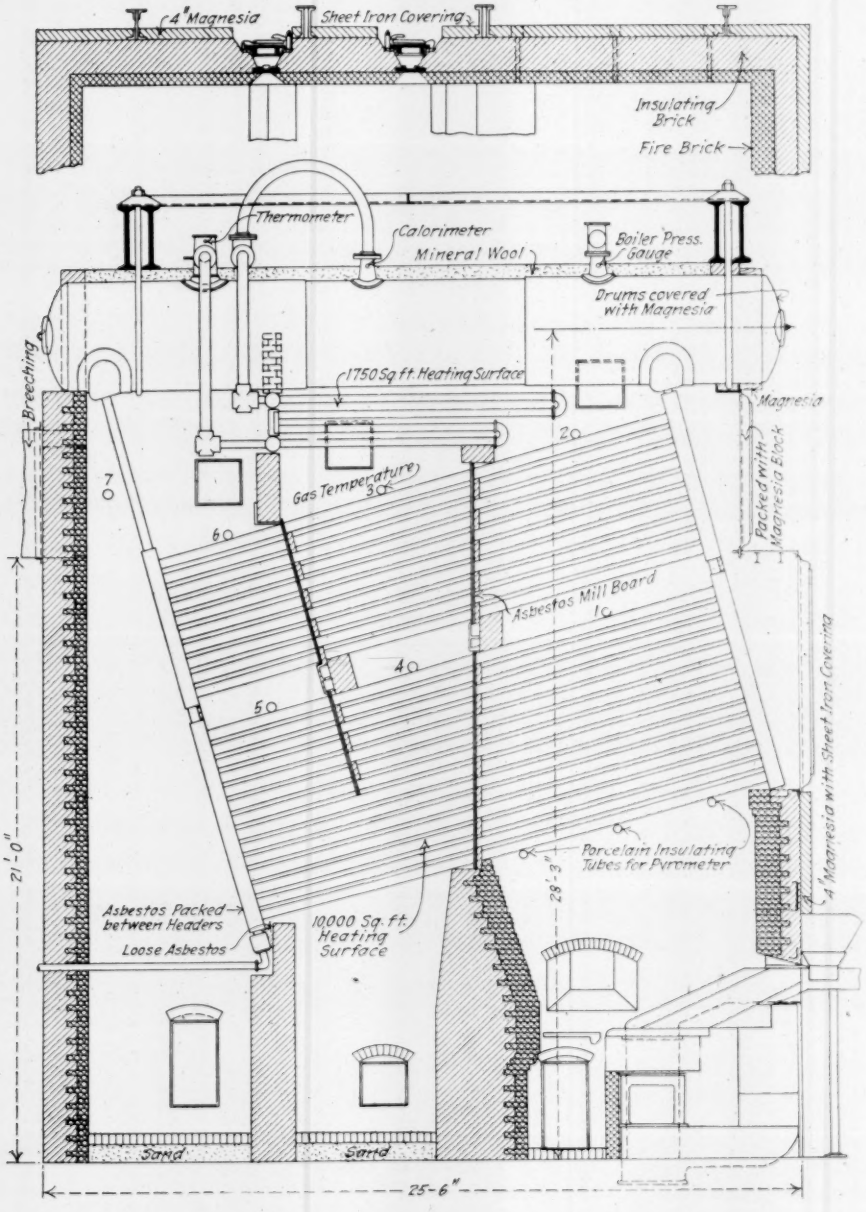

FIG. 1-a LONGITUDINAL SECTION OF BOILER

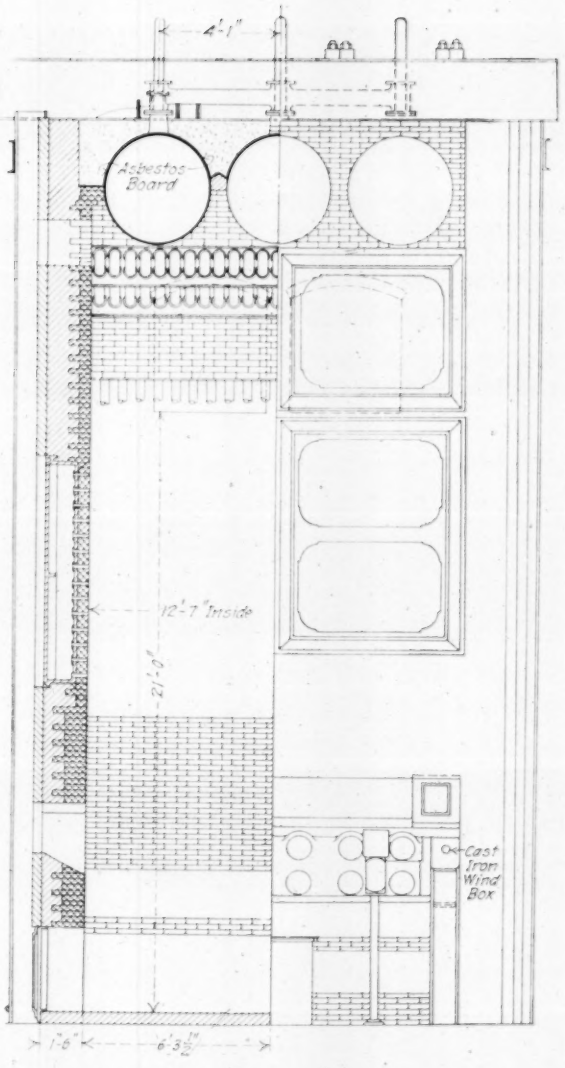

FIG. 1-b FRONT ELEVATION OF BOILER

13 A high reading pyrometer was used for furnace temperatures. There was considerable trouble in keeping this pyrometer in working order and consequently the furnace temperatures are given for only a few of the tests.

14 Samples of gas for analysis were taken in the first pass at the top of the 14th tube, and before the breeching. The samples were taken through six sampling tubes placed in the space between the boiler and back wall, and directly in front of the opening to the breeching. One tube was connected to a CO_2 recorder and the other five were attached to large sampling bottles which were run continuously. The tips of the tubes were so placed that the samples taken through each represented approximately equal volumes passing to the stack.

15 The first 19 tests were made with mixed coal, from four different mines all in the same region. Tests Nos. 21 to 40 were all run on coal from No. 29 mine, lighter than that used on the early tests and giving much less clinker trouble. It made a loose fuel bed and allowed more boiler capacity with less draft in the stoker. For best combined efficiency the dry coal per square foot of grate per hour is about 26 lb., or nearly 270 lb. per retort. The quantity of coal burned per square foot of grate per hour was increased to nearly 63 lb. with a decrease in combined efficiency of 5.5 per cent.

16 To determine the carbon in the refuse the total refuse was spread out upon a large floor, broken up, thoroughly mixed and quartered. The quarter sample was ground, and this ground portion was in turn quartered to get the sample for chemical analysis. The calorific values of the coals were determined by the oxygen-bomb calorimeter.

17 The radiation and unaccounted for losses have been calculated for those tests, for which there were ultimate analyses of the coal. The tests were made during extremely cold weather, and there was over the boiler only a temporary building affording but little protection from the elements; the boiler room temperatures were so low at times the boiler pressure gage piping and the water column blow-off piping froze. In view of these facts and the drafty condition of the boiler room the radiation and unaccounted for losses of from 5 to 7 per cent are worthy of note.

18 The theoretical efficiency was calculated on the assumption that the only heat loss was that to the stack when the combus-

tion was complete without excess air, and the temperature of gas leaving the boiler was equal to the temperature of the wet steam in the boiler. The efficiencies in Table 1 have been calculated for those tests for which there were ultimate analyses of the coal.

19 It may be possible under favorable conditions to reduce the radiation losses to 2.5 per cent. With thorough mixing and a suitable type of combustion chamber the heat losses in the gas may be expected to be not more than 1.5 per cent greater than the theoretical gas loss; the maximum combined efficiency which may be attained under these most favorable conditions is about 85 per cent.

TABLE 1 THEORETICAL EFFICIENCIES

No. of Tests	Theoretical Efficiency, Per Cent	Combined Efficiency, Per Cent	Gain in Efficiency necessary to attain Theoretical Efficiency
8	90.10	79.60	10.50
11	90.15	78.14	12.01
16	90.05	76.18	13.87
18	89.95	75.20	14.75
25	89.52	75.79	13.73
35	89.90	78.33	11.57
39	89.85	78.65	11.20

SUMMARY OF RESULTS

20 The best combined efficiencies are obtained with 56 to 66 per cent of the boiler rating. The efficiencies fall off slowly as the quantity of steam generated is increased. The extreme variation in efficiency shown by the individual tests is from 75.2 to 81.3 per cent.

21 The gain in efficiency due to the superheater seems to bear no definite relation to the amount of moisture in the steam entering the superheater. By gain in efficiency due to superheater is meant the heat absorbed by the superheater in percentage of the total heat in the coal. The superheat shows a general tendency upward with the increase in boiler rating. For 60 per cent of the boiler rating the superheat is about 36 deg. fahr., and for 120 per cent rating about 74 deg. fahr. At 60 per cent rating the weight of steam passing through the super-

heat is 18,000 lb. per hour, the amount for which the superheater was built. The pressure drop through the superheater for its rated quantity of steam is 5 lb.; at 120 per cent boiler rating the weight of steam passing through the superheater is double that for which the superheater was designed, and the pressure drop through superheater is 28 lb.

22 At the low boiler ratings the last pass of the boiler, and in fact the last two passes, are of very little use. The best combined efficiencies were obtained when running between 56 and 66 per cent of boiler rating. At these low ratings the drop in gas temperature through the second and third passes altogether is about 20 deg. fahr., and the heat absorbed is very small, about 1.3 per cent of the total. These two passes have 50 per cent of the boiler heating surface. If the second and third passes were dropped off entirely, the loss in combined efficiency would be about 1 per cent at the low rating. It is only when the boiler has reached 75 per cent rating or more that the gain in economy in the last two passes is sufficient to give a reasonable return upon the investment in heating surface. While the exit gas temperatures were reduced almost to the lowest theoretical limit which can be reached without the use of an economizer, it was done at a large expenditure in heating surface.

23 When running at about 50 per cent of boiler rating the temperature of the gases leaving the boiler is practically that of the steam in the boiler. As the capacity is increased the difference between the temperature of the gases leaving the boiler and the temperature of the steam in the boiler increases. The increase in the difference between these two temperatures seems to be approximately in the same ratio as the increase in capacity, so that for an increase from 50 to 100 per cent of the boiler rating the difference in temperature between exit gases and steam increases about 50 deg. fahr.

24 An examination of the boiler after more than six months' service showed no evidence of pitting of the heating surface of the last pass due to the sulphur content of the coal and low exit gas temperature.

APPENDIX

RATING VS. COMBINED EFFICIENCY

25 Fig. 2, showing the relation between boiler rating and combined efficiency, indicates that the efficiency decreases slowly as the steam generated is increased. The line shows an efficiency of 80 per cent at 50 per cent of rating, and an efficiency of 73.5 per cent at 150 per cent rating, or a decrease of 6.5 per cent in the combined efficiency for an increase of 100 per cent in rating. This decrease in efficiency with an increase in rating seems to be due primarily to the temperature of the gases leaving the boiler.

EFFICIENCY AND RATING VS. TEMPERATURE OF GASES LEAVING BOILER

26 It may be seen from Fig. 2 that the temperature of the gases leaving the boiler goes up with the rating; the increase being about 1 deg. fahr. for 1 per cent rating. On the other hand an increase in the exit gas temperature gives a decrease in the efficiency. As an increase in exit gas temperature gives an increased rating and a decreased efficiency it follows that the efficiency will fall off as the rating increases.

27 The superheat-rating curve shows that the superheat has a general tendency upward with the rating.

28 The line for rating coal per retort per hour shows that the coal consumption increases at a more rapid rate than the steam generated.

GAS TEMPERATURES

29 The gas temperatures throughout the boiler from furnace to breeching, and the percentages of total heating surface (boiler plus superheater) between the several pyrometer positions are given in the tables, and have been plotted also. While these percentages vary irregularly, and seem decidedly freakish in individual cases, the general tendency is to show a decrease in the percentage of heat absorption in the first pass with an increase in boiler rating. In the second and third passes the opposite condition prevails, the percentage of heat absorption increasing with boiler rating.

30 A study of the gas temperature data will explain these conditions. At the low boiler rating the gas temperatures are so low by the time the gases reach the second pass that the rate of absorption is low and the percentage small. In general the temperature of the gas entering the second pass increases with the boiler rating; this increases the rate of heat absorption, and gives to this portion of the boiler a greater proportion of the total work.

TABLE 2 1000-H.P. BOILER, RESUMÉ OF PRINCIPAL RESULTS

Test No.	Per Cent Rating	Combined Efficiency	Temperature of Gases leaving Boiler, Deg. Fahr.	CO_2 in Gases leaving Boiler, Per Cent	Per Cent Combustible in Dry Refuse	B.t.u. per Lb. Dry Coal	Evaporation from and at 212 Deg. Fahr. per Lb. Dry Coal Combined Unit	Quality of Steam leaving Boiler	Radiation and Unaccounted for Losses
1	84.79	78.02	390	10.32	15.49	14334	11.525	97.92	....
3	56.72	80.89	363	.....	10.30	14541	12.121	97.83	....
4	61.77	77.93	368	10.10	10.09	14517	11.658	97.93	....
5	59.48	79.10	361	9.39	13.91	14498	11.818	97.99	....
6	61.43	80.76	368	9.00	8.64	14499	12.067	98.16	....
7	60.87	81.00	367	9.00	1.44	14526	12.125	98.20	....
8	60.60	79.60	367	10.38	3.90	14493	11.889	98.11	5.46
9	57.36	77.71	364	10.16	3.39	14445	11.567	98.00	....
10	55.88	81.30	362	10.02	9.69	14289	11.972	97.97	....
11	67.94	78.14	369	10.30	10.93	14624	11.775	97.94	6.42
12	66.43	80.69	373	11.20	17.91	14454	12.018	98.11	....
13	68.86	78.89	375	10.50	18.08	14396	11.704	98.01	....
14	74.25	78.64	381	10.59	14.75	14602	11.833	98.23	....
15	84.94	76.88	390	10.44	23.79	14467	11.461	98.15	....
16	76.21	76.18	383	10.54	23.52	14590	11.453	98.14	6.99
17	76.77	78.76	383	10.68	21.10	14618	11.864	98.20	....
18	89.08	75.20	401	10.22	20.05	14645	11.349	98.00	7.15
19	90.75	75.42	396	10.77	24.50	14471	11.247	98.01	....
21	108.83	76.21	412	10.75	28.73	14518	11.401	98.03	....
22	106.90	75.50	415	10.89	21.48	14462	11.252	98.02	....
23	93.93	76.18	399	10.15	17.80	14263	11.197	97.84	....
24	107.66	76.87	412	10.00	25.76	14522	11.504	98.00	....
25	127.70	75.79	423	11.66	30.68	14415	11.258	97.99	5.65
26	121.36	75.93	426	11.35	29.18	14290	11.181	98.42	....
27	75.65	77.28	379	10.17	14.94	14284	11.375	98.05	....
29	71.63	78.76	367	10.01	14.37	14254	11.569	98.10	....
30	73.75	77.49	369	9.84	16.23	14430	11.523	98.11	....
33	77.80	76.48	376	9.24	20.14	14380	11.333	98.09	....
34	67.53	78.77	374	9.31	7.99	14376	11.670	98.05	....
35	63.05	78.33	376	9.39	7.07	14455	11.668	98.04	5.00
37	62.46	79.08	369	9.42	7.59	14409	11.742	98.10	....
39	53.98	78.65	360	8.84	4.69	14515	11.764	97.80	4.75
40	53.72	78.79	360	9.43	5.09	14378	11.675	97.93	....

Tests 1–19 inclusive with mixed coal from mines Nos. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

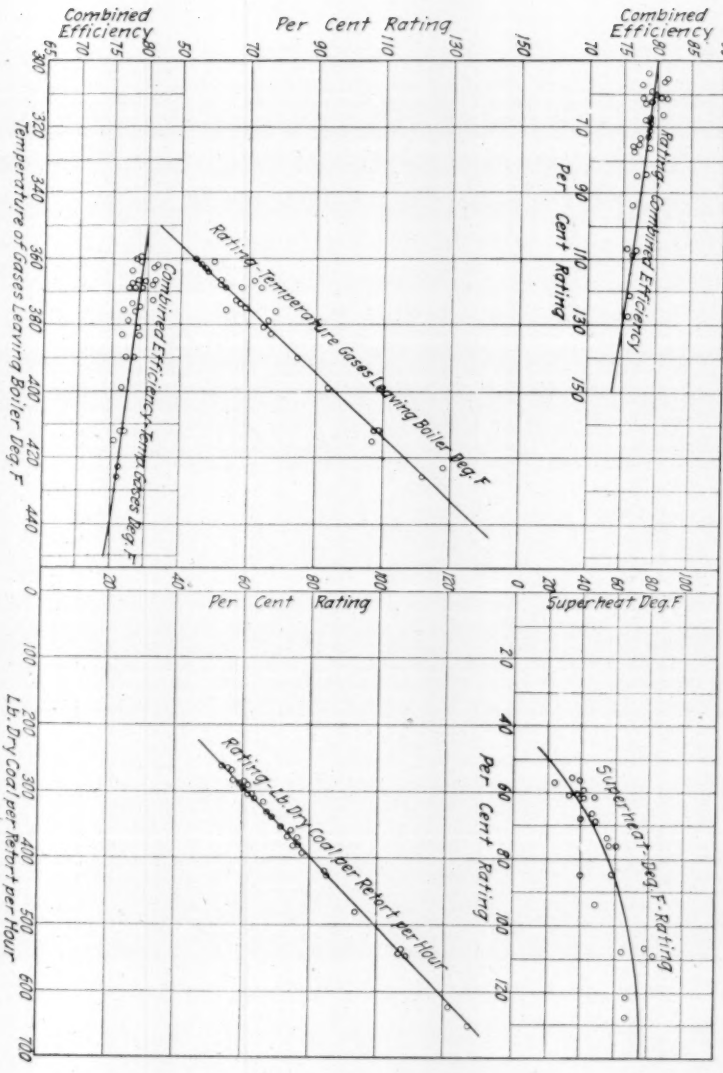
Combined Efficiency
Per Cent Rating
Combined Efficiency
Rating-Combined Efficiency
Per Cent Rating
Rating-Temperature Gases Leaving Boiler Deg. F
Combined Efficiency-Temp Gases Deg. F
Temperature of Gases Leaving Boiler Deg. F
Per Cent Rating
Superheat Deg. F
Superheat Deg. F-Rating
Per Cent Rating
Rating-Lb Dry Coal per Retort per Hour
Lb. Dry Coal per Retort per Hour

FIG. 2 CURVES SHOWING RELATION BETWEEN BOILER RATING AND COMBINED EFFICIENCY

TABLE 3 1000-H.P. BOILER, TEST DATA

Test No.	Date of Test	Length of Test, Hours	Steam Pressure by Gage Lb. per Sq. In.		Average Draft, In. of Water			Temperature, Deg. Fahr						
			In Boiler	Leaving Superheater	Barometer, Lb.	Pressure in Stoker Wind Box	Suction over Fire	Suction before Breeching	Fireroom	Saturated Steam at Pressure, leaving Superheater	Steam leaving Superheater	Superheat	Feedwater entering Boiler	Gases leaving Boiler	Saturated Steam at Boiler Pressure
1	11-16-17-18-11	47.983	136.00	124.00	14.53	4.89	0.157	.....	41.9	352.26	392.00	39.74	147.92	390	358.77
3	11-21-22-11	24.28	127.00	124.00	14.57	2.480	0.148	.....	32.0	352.29	392.00	39.71	133.34	363	353.95
4	11-23-24-11	24.61	130.00	124.00	14.58	2.599	0.144	.....	44.6	352.27	393.80	41.53	142.34	368	355.59
5	11-24-25-11	21.61	126.00	121.00	14.57	2.599	0.146	.....	32.0	350.58	392.00	41.42	141.26	361	353.39
6	11-27-28-11	17.90	131.00	125.00	14.61	2.520	0.137	.....	48.2	352.85	401.00	48.15	146.12	368	356.11
7	11-28-29-11	21.50	130.00	125.00	14.54	2.599	0.137	.....	47.75	352.81	390.20	37.39	144.68	367	355.57
8	12-1-2-11	28.083	131.00	126.00	13.51	2.717	0.104	.....	38.66	354.81	387.50	32.69	142.16	367	355.55
9	12-4-5-11	24.33	126.00	123.00	14.75	2.402	0.098	.....	39.56	351.84	375.80	23.96	125.24	364	353.48
10	12-5-6-11	24.68	128.00	124.00	14.74	2.402	0.137	.....	40.64	352.36	386.60	34.24	125.60	362	354.58
11	12-6-7-11	21.66	141.00	128.00	14.74	3.228	0.126	.....	45.50	354.56	393.80	39.24	131.72	369	361.44
12	12-7-8-11	20.76	136.00	128.00	14.77	3.386	0.122	.....	46.04	354.58	401.00	46.42	129.74	373	358.90
13	12-8-9-11	24.30	136.00	127.00	14.89	3.662	0.137	.....	46.22	354.13	402.80	48.67	125.24	375	358.95
14	12-11-12-11	24.38	135.00	124.00	14.69	3.465	0.110	.....	59.72	352.33	408.20	55.87	145.04	381	358.30
15	12-12-13-11	23.76	134.00	126.00	14.63	4.370	0.118	.....	43.16	353.41	411.80	58.39	139.64	390	357.73

TABLE 3—*Continued*

16	12-13-14-11	18.68	135.00	129.00	14.71	3.110	0.098	.....	39.56	355.12	411.80	56.68	137.48	383	358.32
17	12-14-15-11	23.93	134.00	127.00	14.69	3.465	0.091	0.196	40.10	354.00	415.40	61.40	157.46	383	357.42
18	12-19-20-11	29.11	137.00	126.00	14.84	4.567	0.118	0.359	34.88	353.52	410.00	56.48	137.66	401	359.42
19	12-21-22-11	24.11	140.00	127.00	14.72	5.276	0.137	0.394	36.50	354.02	408.20	54.18	130.64	396	360.86
21	12-26-27-11	22.13	147.00	130.00	14.30	5.040	0.085	0.472	43.88	355.44	438.80	83.36	131.18	412	364.24
22	12-27-28-11	23.70	143.60	127.00	14.42	4.764	0.098	0.484	36.50	353.84	433.40	79.56	119.84	415	362.70
23	12-28-29-11	20.70	144.00	127.00	14.65	4.685	0.173	0.496	40.64	353.98	402.80	48.82	125.42	399	362.93
24	12-29-30-11	23.55	145.00	128.00	14.78	4.488	0.122	0.500	37.94	354.58	419.00	64.42	145.22	412	363.49
25	1-4-5-12	18.00	158.00	129.00	14.53	5.630	0.085	0.461	25.16	355.01	422.60	67.59	141.80	423	369.66
26	1-5-6-12	23.96	150.00	123.00	14.61	5.906	0.075	0.512	12.20	351.77	419.00	67.23	140.90	426	365.85
27	1-18-19-12	22.28	141.00	134.00	14.56	3.071	0.158	0.276	43.70	357.67	404.60	46.93	134.06	379	361.35
29	1-22-23-12	23.95	138.00	129.00	14.57	3.110	0.110	0.173	29.30	355.05	388.40	33.35	139.10	367	359.80
30	1-23-24-12	22.36	137.00	129.00	14.49	3.189	0.106	0.268	31.64	354.99	395.60	40.61	139.10	369	359.26
33	1-26-27-12	24.15	139.00	131.00	14.58	3.583	0.126	0.389	12.56	356.09	392.00	35.91	138.92	376	360.28
34	1-29-30-12	24.016	138.00	132.00	14.53	2.717	0.110	0.358	29.48	356.61	392.00	35.39	146.30	374	359.76
35	1-30-31-12	20.40	134.00	128.50	14.46	2.638	0.126	0.354	29.30	354.69	381.20	26.51	157.28	376	357.64
37	2-1-2-12	25.26	134.00	128.00	14.56	2.795	0.110	0.252	22.46	354.47	377.60	23.13	152.24	369	357.67
39	2-6-7-12	24.53	132.00	127.00	14.60	2.126	0.079	0.126	26.24	353.95	377.60	23.65	157.28	360	356.67
40	2-7-8-12	18.28	135.00	130.00	14.55	2.205	0.067	0.078	26.42	355.57	368.60	13.03	160.70	360	358.23

Tests 1–19 inclusive with mixed coals from mines Nos. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 4 100-H.P. BOILER, TEST RESULTS

Test No.	Relative Humidity of Atmosphere, per Cent	Feedwater, Lb. per Hr.	Evaporation Factor Combined Unit	Evaporation Factor Boiler only	Lb. Dry Coal per Hr.	Lb. Dry Refuse per Hr.	Lb. Combustible assending from Grate per Hr.	Lb. Dry Coal per Hr. per Sq. Ft. Grate.	Lb. Dry Coal per Hr. per Retort	Equivalent Evaporation per Lb. Dry Coal Combined Unit	Equivalent Evaporation per Lb. Dry Coal Boiler only	Per Cent Builders Rating Developed	Efficiencies: Boiler, Furnace, and Superheater	Efficiencies: Boiler and Furnace	Efficiencies: Boiler Only	Gain in Efficiency Due Superheater
1	....	25814	1.13317	1.09201	2538	263	2275	40.63	423.00	11.525	11.106	84.79	78.02	75.18	75.20	2.84
3	....	17045	1.14817	1.10516	1615	144	1471	25.85	269.17	12.121	11.666	86.72	80.89	77.85	77.54	3.04
4	....	18692	1.13998	1.09712	1828	150	1678	29.26	304.67	11.658	11.219	61.77	77.93	74.99	74.62	2.94
5	....	17994	1.14048	1.09829	1736	158	1579	27.79	289.33	11.818	11.381	59.48	79.10	76.18	76.25	2.92
6	....	18590	1.14004	1.09539	1756	154	1603	28.11	292.67	12.067	11.594	61.43	80.76	77.59	77.43	3.17
7	....	18487	1.13592	1.09710	1732	138	1594	27.73	288.67	12.125	11.711	60.87	81.00	78.23	77.43	2.77
8	....	18400	1.13632	1.09886	1759	143	1616	28.16	293.17	11.889	11.498	60.60	79.60	76.99	76.30	2.61
9	....	17253	1.14706	1.11486	1711	138	1573	27.39	285.17	11.567	11.243	57.36	77.71	75.53	74.71	2.18
10	....	16719	1.15298	1.114427	1610	158	1453	25.77	268.33	11.972	11.752	55.88	81.30	79.81	78.17	1.49
11	....	20366	1.15026	1.10953	1989	158	1831	31.84	331.50	11.775	11.358	67.94	78.14	75.37	75.22	2.77
12	....	19817	1.15645	1.11249	1907	184	1723	30.53	317.83	12.018	11.561	66.43	80.69	77.61	78.05	3.08
13	....	20440	1.16219	1.11622	2030	206	1824	32.50	338.30	11.704	11.241	68.86	78.89	75.77	76.23	3.12
14	....	22365	1.14537	1.09763	2165	181	1984	34.66	360.80	11.833	11.339	74.25	78.64	75.36	75.40	3.28
15	....	25424	1.15264	1.10235	2557	246	2311	40.93	426.20	11.461	10.961	84.94	76.88	73.52	74.16	3.36

TABLE 4—*Continued*

16	....	22776	1.15445	1.10462	2296	209	2087	36.76	382.66	11.453	10.958	76.21	76.18	72.88	73.47	3.30
17	....	23311	1.13620	1.08445	2233	189	2044	35.75	372.17	11.864	11.323	76.77	78.76	75.17	75.67	3.59
18	....	26640	1.15365	1.10354	2708	243	2465	43.35	451.33	11.349	10.857	89.08	75.20	71.94	72.45	3.26
19	53.0	26997	1.15974	1.11113	2784	261	2523	44.57	464.00	11.247	10.776	90.75	75.42	72.26	73.20	3.16
21	53.5	31933	1.17576	1.11153	3293	339	2954	52.72	548.80	11.401	10.778	108.83	76.21	72.04	73.50	4.17
22	49.5	31128	1.18480	1.12276	3278	333	2945	52.48	546.30	11.252	10.663	106.90	75.50	71.55	72.50	3.95
23	55.1	27889	1.16200	1.11551	2894	315	2579	46.33	482.30	11.197	10.749	93.93	76.18	73.13	73.70	3.05
24	55.3	32282	1.15060	1.09658	3229	318	2911	51.69	538.20	11.504	10.964	107.66	76.87	73.26	74.29	3.61
25	51.3	38110	1.15605	1.10142	3913	443	3470	62.64	652.20	11.258	10.726	127.70	75.79	72.21	74.02	3.58
26	53.4	36225	1.15576	1.10538	3744	452	3292	59.94	624.00	11.181	10.694	121.36	75.93	72.62	74.23	3.31
27	51.0	22631	1.15327	1.10808	2294	234	2060	36.72	382.30	11.375	10.929	75.65	77.28	74.25	74.36	3.03
29	53.0	21690	1.13932	1.10299	2136	224	1912	34.20	356.00	11.569	11.200	71.63	78.76	76.25	76.43	2.51
30	51.9	22249	1.14357	1.10299	2208	215	1993	35.35	368.00	11.523	11.114	73.75	77.49	74.74	74.94	2.75
33	51.7	23520	1.14127	1.10314	2369	237	2131	37.93	394.80	11.333	10.954	77.80	76.48	73.92	74.37	2.56
34	54.0	20553	1.13357	1.09515	1996	183	1813	31.95	332.70	11.670	11.275	67.53	78.77	76.11	75.71	2.66
35	53.0	19485	1.11639	1.08318	1864	157	1707	29.84	310.70	11.668	11.321	63.05	78.33	76.00	75.55	2.33
37	52.3	19249	1.11952	1.08892	1835	162	1673	29.38	305.80	11.742	11.421	62.46	79.08	76.92	76.61	2.16
39	49.4	16710	1.11448	1.08081	1583	124	1459	25.34	263.80	11.764	11.409	53.98	78.65	76.27	75.66	2.38
40	50.0	16775	1.10487	1.07883	1587	138	1449	25.41	264.50	11.675	11.400	53.72	78.79	76.94	76.33	1.85

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 5 1000-H.P. BOILER, TEMPERATURE AND DRAFTS

Test No.	Furnace	Gas Temperatures, Deg. Fahr.							Draft in Stoker and Wind Boxes, In. Water						
		No. 1 In First Pass over 14th Tube	No. 2 In First Pass over 24th Tube	No. 3 In Second Pass over 24th Tube	No. 4 In Second Pass over 14th Tube	No. 5 In Third Pass over 14th Tube	No. 6 In Third Pass over 24th Tube	No. 7 Leaving Boiler between Boiler and Back Wall)	East Tuyere Box	West Tuyere Box	Bottom West Outside Box	Bottom East Outside Box	Center or Main Wind Box	Top East Outside Box	Top West Outside Box
1	....	759	572	460	423	388	385	390	4.01	4.13	4.76	5.19	4.89	6.14	5.98
3	....	691	518	387	399	378	378	363	2.16	2.20	2.87	2.95	2.48	4.09	4.01
4	....	631	498	406	361	365	363	368	2.32	2.40	2.91	3.18	2.59	4.05	3.89
5	....	639	509	406	365	367	370	361	2.32	2.40	2.91	3.11	2.59	4.05	3.97
6	....	666	500	408	390	378	367	368	2.32	2.32	2.91	3.07	2.51	4.01	3.97
7	....	639	482	387	381	370	363	367	2.36	2.36	2.95	3.11	2.59	3.97	3.93
8	1763	658	487	385	370	370	369	367	2.44	2.51	2.99	3.14	2.71	4.05	3.97
9	1841	626	417	365	352	356	363	364	2.16	2.24	2.79	2.87	2.40	3.89	3.81
10	1800	644	493	379	365	363	370	362	2.24	2.32	2.83	2.87	2.40	3.93	3.85
11	1888	696	505	387	383	363	372	369	2.95	3.07	3.50	3.66	3.22	4.60	4.52
12	1866	738	532	396	392	370	379	373	3.18	3.22	3.66	3.81	3.38	4.68	4.60
13	1958	801	572	428	414	385	390	375	3.42	3.50	3.85	4.09	3.66	4.68	4.60
14	1902	738	554	424	408	379	381	381	3.18	3.22	3.66	3.97	3.46	4.60	4.52
15	1989	838	621	477	441	396	401	390	4.05	4.17	4.76	5.07	4.37	6.10	5.94

TABLE 5—*Continued*

16	....	727	538	423	390	378	374	383	2.83	2.95	3.50	3.74	3.11	4.68	4.60
17	....	788	594	433	408	385	385	383	3.14	3.30	3.70	3.97	3.46	4.68	4.60
18	....	851	651	496	394	405	403	401	4.13	4.29	4.76	5.19	4.56	5.90	5.82
19	....	810	612	518	441	401	397	396	4.80	5.00	5.35	5.70	5.27	6.29	6.18
21	2176	900	698	603	495	426	419	412	4.52	4.68	5.07	5.51	5.03	6.10	6.02
22	....	914	712	604	491	424	414	415	4.17	4.33	4.84	5.39	4.76	6.02	5.90
23	....	838	617	507	459	401	397	399	4.13	4.21	4.56	4.96	4.68	5.51	5.39
24	....	867	653	525	468	412	405	412	3.93	4.09	4.48	4.92	4.48	5.51	5.39
25	....	961	711	541	491	426	423	423	4.92	5.00	5.59	6.10	5.63	6.85	6.73
26	....	982	752	563	504	438	437	426	5.35	5.43	5.98	6.45	5.90	7.08	6.88
27	....	819	610	478	439	401	387	379	2.71	2.83	3.03	3.42	3.07	3.62	3.62
29	....	689	570	423	401	385	383	367	2.71	2.87	3.18	3.66	3.11	4.05	3.97
30	....	703	612	471	435	396	396	369	2.83	2.91	3.18	3.58	3.18	3.81	3.74
33	....	792	599	464	428	399	388	376	3.18	3.26	3.54	3.93	3.58	4.25	4.21
34	....	716	550	455	414	381	379	374	2.44	2.55	2.71	2.95	2.71	3.14	3.11
35	....	714	549	453	408	381	378	376	2.28	2.40	2.59	2.83	2.63	3.11	3.07
37	....	727	545	442	405	379	377	369	2.48	2.51	2.75	2.99	2.79	3.18	3.18
39	....	649	496	397	388	367	363	360	1.92	2.04	2.24	2.48	2.12	2.83	2.79
40	....	603	446	370	367	352	354	360	1.96	2.12	2.24	2.40	2.20	2.63	2.63

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 6 1000-H.P. BOILER, GAS ANALYSIS

Test No.	Analysis of Gas in First Pass. Top of 14th Tube			Analysis of Gas before Breeching. Gas Leaving Boiler		
	Per Cent CO_2	Per Cent O	Per Cent CO	Per Cent CO_2	Per Cent O	Per Cent CO
1	11.70	5.42	0.160	10.32	7.44	0.035
3	10.70	6.60	0.166	No record		
4	10.22	5.35	0.040	10.10	Machine record	
5	10.47	5.83	0.070	9.39	Machine record	
6	10.15	7.25	0.050	9.00	Machine record	
7	10.30	7.53	0.000	9.00	Machine record	
8	10.64	7.54	0.064	10.38	8.17	0.030
9	10.62	7.88	0.000	10.16	8.41	0.025
10	11.08	7.85	0.058	10.02	8.60	0.000
11	10.90	7.66	0.030	10.30	Machine	
12	12.13	6.37	0.040	11.20	Machine	
13	12.28	4.98	0.250	10.50	Machine	
14	12.09	6.34	0.070	10.59	7.97	0.012
15	12.21	6.05	0.020	10.44	8.00	0.002
16	11.75	6.42	0.000	10.54	8.06	0.057
17	12.02	6.29	0.100	10.68	7.73	0.046
18	11.42	7.40	0.020	10.22	8.52	0.004
19	12.19	5.94	0.067	10.77	7.86	0.022
21	11.89	6.69	0.000	10.75	8.22	0.000
22	12.59	6.13	0.071	10.89	7.87	0.059
23	11.40	7.31	0.057	10.15	8.44	0.029
24	11.91	6.61	0.022	10.00	8.47	0.000
25	13.70	4.30	0.016	11.66	6.03	0.0098
26	13.35	4.60	0.025	11.35	6.78	0.000
27	11.67	7.00	0.000	10.17	8.30	0.000
29	11.07	7.63	0.000	10.01	8.65	0.000
30	11.43	7.07	0.033	9.84	8.83	0.011
33	10.12	8.68	0.000	9.24	9.56	0.000
34	9.91	8.99	0.090	9.31	9.44	0.026
35	10.81	7.84	0.000	9.39	9.45	0.000
37	10.60	8.12	0.000	9.42	9.15	0.000
39	10.39	8.22	0.000	8.84	9.83	0.000
40	10.69	7.96	0.000	9.43	9.43	0.000

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 7 1000-H.P. BOILER. COAL ANALYSIS. PERCENTAGES BY WEIGHT CALORIFIC VALUES

Test No.	Proximate Analysis Dry Coal			Moisture in Coal as Fired	Carbon in Dry Refuse, per Cent	B.t.u. per Lb. Dry Coal	B.t.u. per Lb. Combustible.	Per Cent Sulphur in Dry Coal
	Volatile Matter	Fixed Carbon	Ash					
1	28.99	62.00	9.01	2.88	15.49	14334	15987	2.59
3	29.50	62.53	7.97	3.20	10.30	14541	16031	2.94
4	29.97	62.53	7.50	3.26	10.09	14517	15894	2.40
5	29.74	62.51	7.75	3.04	13.91	14498	15926	2.62
6	28.18	63.90	7.92	2.97	8.64	14499	15922	1.83
7	28.86	63.31	7.83	3.78	1.44	14526	15945	1.92
8	28.85	63.39	7.76	4.29	3.90	14493	15913	2.49
9	29.22	62.97	7.81	2.97	3.39	14445	15881	2.57
10	29.24	61.76	9.00	2.60	9.69	14289	15922	2.57
11	29.63	63.29	7.08	2.95	10.93	14624	15922	2.44
12	29.22	62.76	8.02	2.97	17.91	14454	15910	2.24
13	29.70	62.03	8.27	3.29	18.08	14396	15922	3.03
14	29.78	62.98	7.24	2.80	14.75	14602	15922	2.20
15	29.98	62.35	7.67	3.11	23.79	14467	15869	2.32
16	29.79	63.00	7.21	3.41	23.52	14590	15922	2.15
17	30.20	63.00	6.80	3.74	21.10	14618	15860	2.07
18	29.42	63.37	7.21	3.40	20.05	14645	15971	2.29
19	30.57	62.34	7.09	3.15	24.50	14471	15761	2.31
21	28.40	64.13	7.47	3.60	28.73	14518	15864	1.77
22	28.36	63.75	7.89	4.10	21.48	14462	15883	1.83
23	28.31	62.65	9.04	4.90	17.80	14263	15885	1.95
24	28.40	64.05	7.55	3.99	25.76	14522	15883	1.90
25	28.15	63.76	8.09	3.77	30.68	14415	15858	1.56
26	28.30	62.70	9.00	4.15	29.18	14290	15903	1.83
27	27.86	63.20	8.94	4.27	14.94	14284	15883	1.88
29	28.12	62.78	9.10	4.73	14.37	14254	15885	2.06
30	28.20	63.45	8.35	3.76	16.23	14430	15947	2.18
33	28.56	63.06	8.38	3.70	20.14	14380	15883	1.94
34	28.27	63.21	8.52	4.30	7.99	14376	15915	2.08
35	28.66	63.40	7.94	4.68	7.07	14455	15883	1.91
37	28.11	63.73	8.16	4.61	7.59	14409	15871	1.79
39	28.92	63.57	7.51	4.12	4.69	14515	15883	2.16
40	29.27	62.51	8.22	4.23	5.09	14378	15874	2.35

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

THE V-NOTCH WEIR METHOD OF MEASUREMENT

By D. Robert Yarnall

ABSTRACT OF PAPER

This paper explains the V-notch weir method of measurement and a convenient form of apparatus, the Lea recorder, which can be used for recording graphically and continuously and with a high degree of accuracy rates of flow over V-notches and other weirs. It is shown why the V-notch is the most accurate and convenient form of weir for general testing purposes.

The paper also gives details of tests made with the apparatus measuring hot boiler feedwater at the Dartmouth College power plant during both winter and summer load conditions. The results of these tests show the great accuracy of the V-notch at low rates of flow as well as high.

Results of additional tests recently made in Philadelphia and near New York are given which together with the Dartmouth College tests show that the V-notch method of measurement gives results at high, intermediate and low rates of flow which can be relied upon to within one per cent.

THE V-NOTCH WEIR METHOD OF MEASUREMENT

By D. Robert Yarnall, Philadelphia, Pa.

Member of the Society

Attempts to measure accurately and to record automatically the flow of water through pipes or channels have been made from time to time with varying results as to accuracy and convenience. This paper will attempt to show results obtained by the simple and practical V-notch method of measurement, which though not new, seems now to be meeting with marked success, both abroad and in America, in power plant measuring problems.

2 Two tests of the accuracy of this method of measurement have recently been made in America and the data collected are here presented to show the degree of accuracy attainable.

3 The instrument employed in the tests is the Lea V-notch recorder of which a description follows:

4 This apparatus depends on the laws governing the flow of water through weirs, which laws have been found to work with extraordinary accuracy. It is only comparatively recently, however, that they have been turned to account in measuring widely varying rates of flow, both hot and cold, met with in power plants.

5 The most common form of weirs is the plain horizontal sill and the sharp edge rectangular notch. The former of these has the full width of the stream or channel and is without end contraction, the formula used in connection with it being that of Francis

$$\text{cubic feet per second} = 3.33\ LH\sqrt{H}$$

where

L = the width of the weir in feet.

H = the head of water passing over it in feet.

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6 The second weir is a sharp edge rectangular notch of less width than the channel or stream and therefore has end contraction. The formula used in this is

$$\text{cubic feet per second} = 3.33\, LH\, \sqrt{H \times C}$$

where

L and H are the same as before.

C = a constant which depends on the ratio of the width of the weir to the width of the channel.

7 Although these two formulae give a fair degree of accuracy there is another case in which the accuracy is much greater and approaches very nearly 100 per cent. This is the

TABLE 1 FLOW THROUGH 90-DEG. V-NOTCH WEIRS

Depth in Notch, In.	Flow per Hour, Lb.	Depth in Notch, In.	Flow per Hour, Lb.
1	1,140	9	277,960
2	6,480	10	361,740
3	17,830	11	459,030
4	36,610	12	568,720
5	63,940	13	694,710
6	100,860	14	836,110
7	148,290	15	993,510
8	207,060		

sharp edge V-notch of Prof. James Thomson, the formula of which is

$$\text{cubic feet per minute} = 0.305\, H^{\frac{5}{2}}\, \sqrt{H}$$

where

H = the height of the notch in inches, the angle of the notch being 90 deg.

8 Table 1 gives the flow of water through 90-deg. V-notches for each inch of head, up to and including 15 in., calculated by Thomson's formula.

9 The section of the flow through the V-notch is at all times the same shape, though the area may vary, and this constancy of form tends to simplify the formula and make it accurate. This form of notch is also especially adapted for measuring smaller quantities of water than the rectangular weirs, as shown by the curves in Fig. 1 which give a comparison of the accuracy of the different types of weirs at different rates of flow. Another property of the notch is that its angle may be less than 90 deg.

without impairing its efficiency, which enables it to be used to measure very small quantities of water.

10 It was for these reasons, as well as for its accuracy, that the V-notch was adopted for use in connection with the Lea recorder. A sectional view of the apparatus is shown in Fig. 2. It will be seen that a float spindle, rigidly attached to a float

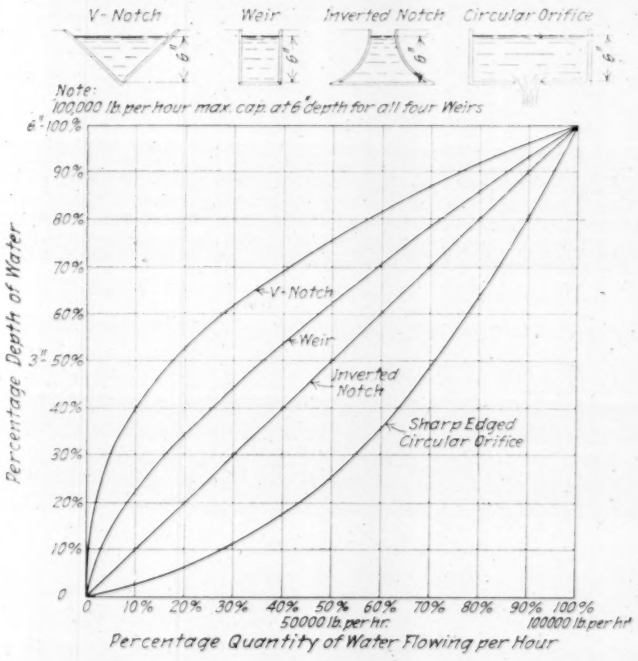


FIG. 1 CURVE SHOWING COMPARATIVE ACCURACY OF DIFFERENT TYPES OF WEIRS UNDER DIFFERENT RATES OF FLOW

which rides on the surface of the water flowing over the V-notch, passes up through the bottom of the instrument case. A rack on this spindle gears into a small pinion upon the axis of a drum revolving between centers.

11 Upon the body of the drum is a screw thread, Fig. 3, the contour of which is the "curve of flow" for the V-notch in connection with which the recorder is used, and as the flow through the notch increases rapidly with the depth, the pitch of the screw increases in the same proportion. Above this drum is a horizontal slider bar, supported on small pivoted rollers and carrying an

arm, at the upper end of which is a pen or pencil point in contact with a paper chart upon a clock driven recording drum, which revolves once in 24 hours (Fig. 4). As the float rises or falls the drum spiral is rotated, and its motion is imparted to the slider bar and pen arm by means of a saddle arm projecting from the latter and engaging at its lower end with the screw

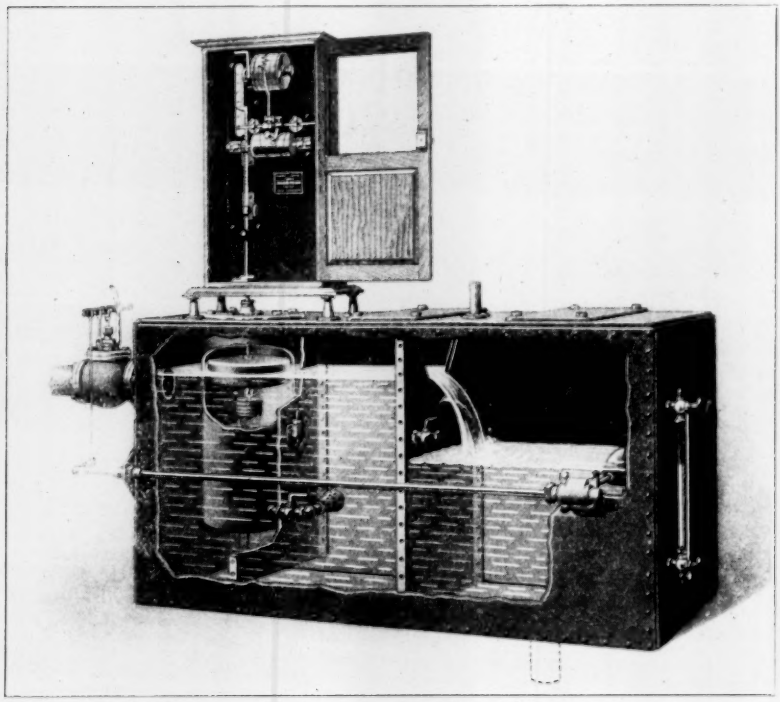


FIG. 2 SECTIONAL VIEW OF APPARATUS USED IN TESTS

thread. On the body of the spiral drum is a graduated scale adjacent to the screw thread and computed in pounds per hour. It is thus seen that the drum serves the double purpose of rectifying the motion of the pen so that it moves equal distances for equal increments in the rate of flow; and of providing a magnified scale for making an accurate observation of the rate of flow at any moment.

12 The apparatus has the following advantages:

a It is simple and hence dependable.

b Its operation is continuous.

c There are no moving parts in contact with the flow except the float, which is in a chamber by itself and not in the path of the flow.

d Collections of sediment and deposits of scale do not affect the accuracy.

e Owing to change of density of the liquid being measured due to changes in temperature, the float which actuates the recorder is immersed more or less, depending upon this change of density, and the recorder and the float are so constructed that they compensate automatically for these changes in temperature over a wide range.

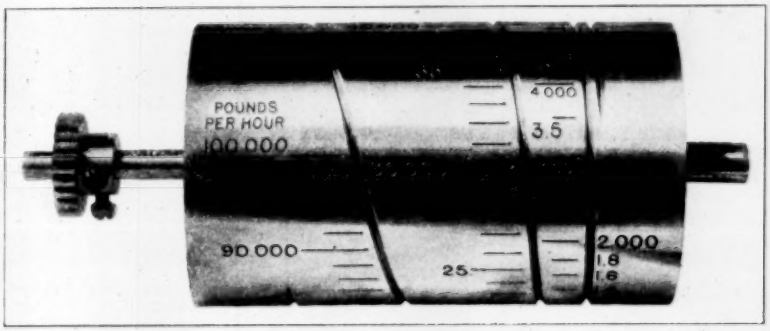


FIG. 3 DETAIL OF RECORDER DRUM SHOWING GRADUATED CURVE OF FLOW

f By making the tank of ample proportions a considerable interval of time takes place between changes in the head on the weir, so that the recording apparatus has a chance to adjust itself to changes in the rate of flow, thus giving a chart very easy to measure with the planimeter.

g It is always placed on suction side of pump, hence it is available at all times for inspection, cleaning and repairs.

13 Like the Venturi meter and the Pitot tube, the V-notch recorder is a continuous flow meter, but unlike these it is not dependent for its accuracy on velocity changes, and hence its record is not subject to the fluctuations due to the sudden changes

of velocity head when these meters are in service, measuring rapidly changing rates of flow.

14 Perhaps the greatest advantage of this method of measurement is found in the way by which it may be checked independently of the recording mechanism. The operator may construct alongside of his recorder an independent "hook" gage or use the graduated vertical scale and pointer on the extended float

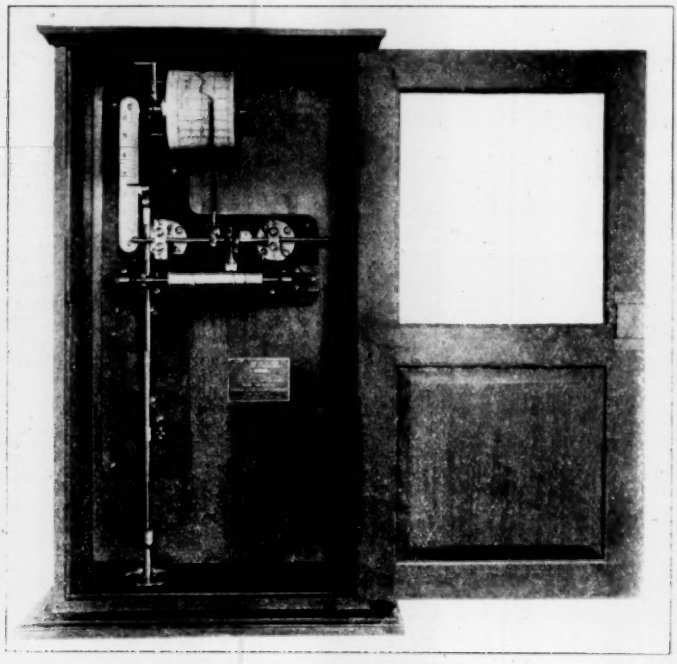


FIG. 4 DETAIL OF RECORDER

rod in the instrument, and can observe from time to time the number of inches head on the weir. By substituting these values in Thomson's formula in Par. 7, the rates of flow as shown on the recorder chart may be checked quickly and accurately. Thus the operator need not depend upon the recorded result until he is sure that it agrees with the figured result from his hook gage readings. The author believes, therefore, that the V-notch with its tank is the most satisfactory testing set that could be devised for measuring large and small rates of flow, hot and cold, as well as fluctuating or uniform conditions of flow.

15 For results the following reports of tests are given, and as clearly as possible the actual test conditions which prevailed explained.

16 With a view to checking the accuracy of this apparatus and the V-notch for the measurement of boiler feedwater a series of tests were made at the Dartmouth College power plant, at both winter load and summer load. These tests were conducted

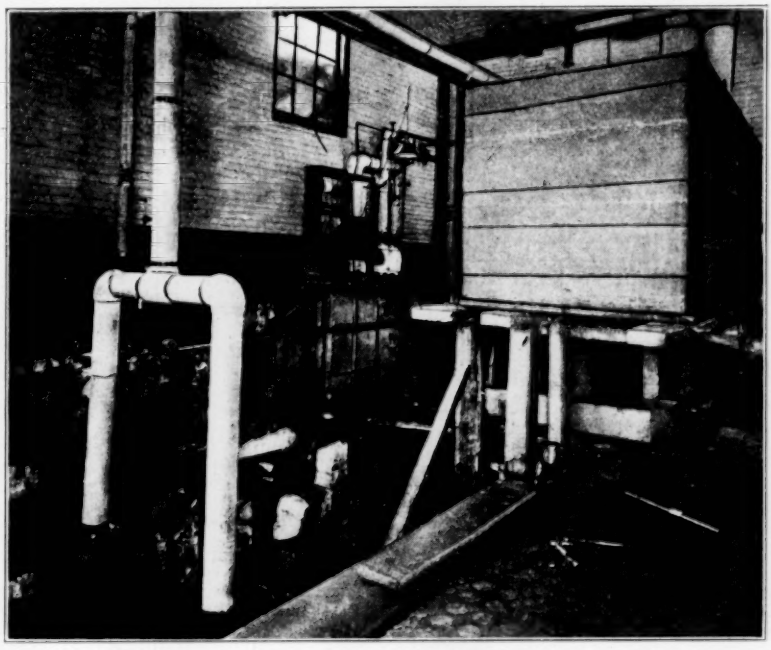


FIG. 5 ARRANGEMENT OF TESTING APPARATUS FOR THE DARTMOUTH COLLEGE TEST

by the engineers for the college, Messrs. R. D. Kimball Company, through their representative J. H. Browne, assisted by the writer.

DARTMOUTH COLLEGE WINTER TEST

17 Fig. 5 is from a photograph of the testing apparatus and meter. On the right-hand side is a large wooden tank divided into two parts by a vertical partition. The size of this tank was 54⅝ in. deep by 54¼ in. wide by one section 59 1/16 in. long and the other section 59⅛ in. long, inside measurements. To the

left is the cast-iron V-notch tank and mounted above is the recording apparatus. The inlet of the meter is controlled by a 6-in. balanced float valve operated by a float riding on the surface of the storage section of the notch tank below the weir, so as to prevent flooding in the event of the feed pump slowing down or stopping (Fig. 2).

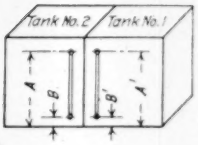
18 The return water from all of the buildings of Dartmouth College is collected in the receiver shown at the left of the meter tank. The make-up water also enters this receiver, which is under about 5 lb. back pressure. The 4 in. line discharges from this receiver into either one of the sections of the wooden measuring tanks, the water flowing by gravity from this tank through the meter to the feed pump and is thence pumped into the boilers in the adjoining boiler plant.

19 The two sections of the wooden tank were provided with independent vertical glass columns ¾ in. in diameter. Just back of and touching each glass column was provided a scale graduated in 16ths of an inch. The two tanks were carefully calibrated by actually weighing the value of 1 in. in height of the water in the tank at a given temperature, this value being independently computed as noted in the tables. It was found during the test that the meniscus of the column at the upper and lower reading could be read with great accuracy because of the careful construction of the column and the graduation of the scale.

20 It is to be noted that this test was made at the normal running load and took care of all of the natural fluctuations in the returns from the heating system. The test was conducted when the outside temperature was only slightly above zero most of the time, hence the heating system was being taxed to its utmost.

21 The test was run under exact working conditions of the plant at heavy load, which fluctuated badly (Fig. 6A). These fluctuations in the load were caused by poor control of the return water from the new gymnasium which was at a lower elevation than the power plant and the accumulated returns from the heating system were intermittently pumped up into the power plant by a float controlled motor driven pump. Only very small water storage was provided in the receiver, hence the fluctuating boiler feed conditions shown: under these conditions the results of the test are remarkably accurate.

TABLE 2 TEST OF V-NOTCH RECORDER AT DARTMOUTH COLLEGE, MARCH 23, 1912



$A - B = C$

$A' - B' = C'$ Correct Quantity $= \Sigma (C + C')$

METER CAPACITY, 50,000 LB. PER HR.; 6-IN. RECORDER; 90° V-NOTCH 6 IN. DEEP

	TANK NO. 1			TANK NO. 2			
Time	A	B	$A-B$	A'	B'	$A'-B'$	Temperature, Deg. Fahr.
9.00	$37\frac{3}{16}$	$7\frac{5}{8}$	$29\frac{3}{16}$	$42\frac{5}{8}$	5	$37\frac{5}{8}$	164
9.22	$42\frac{1}{2}$	$3\frac{5}{8}$	$38\frac{7}{8}$	$34\frac{1}{2}$	3	$31\frac{1}{2}$	
9.41	$32\frac{7}{8}$	$3\frac{1}{8}$	$29\frac{3}{4}$	$39\frac{7}{8}$	$2\frac{3}{4}$	$37\frac{1}{8}$	180
9.50	$41\frac{1}{8}$	$3\frac{1}{4}$	$37\frac{7}{8}$	$42\frac{3}{4}$	$8\frac{3}{4}$	34	
10.15	$43\frac{7}{8}$	$7\frac{1}{8}$	$36\frac{3}{4}$	$43\frac{1}{8}$	$3\frac{3}{8}$	$39\frac{3}{4}$	180
10.40	42	$2\frac{3}{4}$	$39\frac{1}{4}$	$39\frac{7}{8}$	$4\frac{1}{2}$	$35\frac{3}{8}$	
11.06	$40\frac{1}{4}$	2	$38\frac{1}{4}$	$42\frac{3}{4}$	$3\frac{1}{2}$	$39\frac{1}{4}$	170
11.40	44	$3\frac{7}{8}$	$40\frac{1}{8}$	44	$7\frac{1}{4}$	$36\frac{3}{4}$	
12.00	$43\frac{1}{2}$	3	$40\frac{1}{2}$	$43\frac{1}{4}$	3	$40\frac{1}{4}$	160
12.27	44	$5\frac{3}{8}$	$38\frac{5}{8}$	$44\frac{5}{8}$	$2\frac{1}{2}$	$42\frac{1}{8}$	
1.00	44	2	42	44	$1\frac{3}{4}$	$43\frac{1}{4}$	195
1.25	$44\frac{3}{8}$	$2\frac{1}{4}$	$42\frac{1}{8}$	$44\frac{1}{4}$	$2\frac{1}{8}$	$42\frac{1}{8}$	
1.50	$46\frac{1}{4}$	$4\frac{3}{8}$	$41\frac{7}{8}$	$44\frac{1}{2}$	$2\frac{7}{8}$	$41\frac{5}{8}$	195
2.15	43	$2\frac{1}{4}$	$40\frac{3}{4}$	44	$\frac{5}{8}$	$43\frac{3}{8}$	
2.52	$39\frac{3}{8}$	$3\frac{3}{8}$	38	$43\frac{7}{8}$	$2\frac{3}{4}$	$41\frac{1}{8}$	194
3.25	44	$1\frac{5}{8}$	$43\frac{7}{8}$	$41\frac{1}{8}$	$3\frac{1}{2}$	$37\frac{5}{8}$	190
4.00		Total... $615\frac{3}{16}$ in. or 615.18 in.			Total.. $621\frac{7}{8}$ in. or 621.87 in.		

Tank calibration by actual weight at 175 deg. fahr.:

In tank No. 2, 1 in. = 117.0 lb.
In tank No. 1, 1 in. = 115.2 lb.

$615.16 \times 117.0 =$	71976	
$621.87 \times 115.2 =$	71640	
Total.....	143616	
Error due to tank leakage	225	143841

From planimeter measurement of chart (see Fig. 6A)
Quantity, lb. = Area × Constant
$4.04 \times 36312 = 146700.48$

Correction for zero adjustment of recorder.....	1,162	145538
Recorder error (high) =		1697
Recorder error, per cent =		1.18

DARTMOUTH COLLEGE SUMMER TEST

22 The second test was made in the summer to check accurately the claim that the meter is as accurate at low rates of flow as at high. Table 3 substantiates this claim as the accuracy was even greater (0.55 per cent) than that at the winter

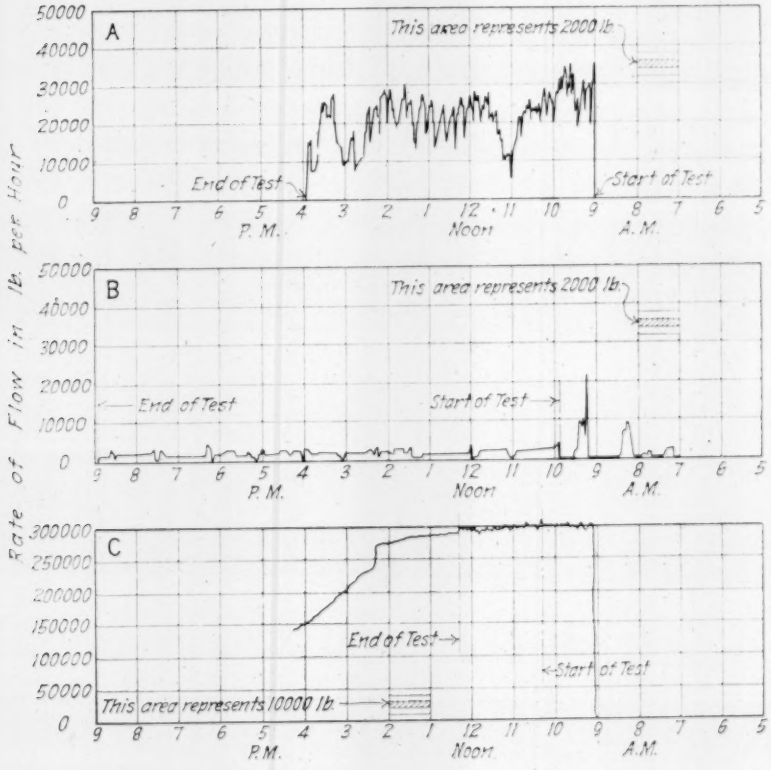


FIG. 6 RECORDING CHARTS

load. This due to the more even rate of flow (see low reading) which in turn is due to the gymnasium heating system not being in service.

TESTS ON V-NOTCH METER IN PHILADELPHIA, AUGUST 9, 1912

23 Figs. 7 and 8 show the arrangements for another test conducted recently in exactly the same way as the tests at Dartmouth

College except that the water was measured on the discharge side of weir instead of the intake side. In this test, however, the water being measured was at 73 deg. while the Dartmouth tests were run at an average of 190 deg. fahr.

24 The V-notch meter was mounted for convenience on a platform car on the coal siding shown in Fig. 8. The car during the test was rigidly jacked to prevent vibration.

TABLE 3 TEST OF V-NOTCH RECORDER AT DARTMOUTH COLLEGE, AUGUST 2, 1912

Tank No. 1				Tank No. 2			
Time	Full, In.	Empty, In.	Weight	Time	Full, In.	Empty, In.	Weight
10.52	32.0	....	.....	9.43	44.81	6.50	4495.3
11.47	....	1.68	3502.3	11.41	32.44	7.81	2890.0
1.03	40.0	0.875	4520.8	2.03	33.81	1.75	3761.9
2.55	34.5	2.00	3655.4	3.54	41.50	1.63	4680.1
4.55	41.25	9.125	3712.0	6.00	45.31	1.19	5177.0
7.15	40.25	7.06	3880.8	8.00	41.38	29.87	1349.4
		Total.....	19271.3				22353.7
							19271.3
						Total.....	41625.0

Chart area = 0.570
Constant = 36312. lb. per sq. in.
Product = 20697.84
As 90 V-notch is used multiply by 2 as drum is cut for $\frac{1}{2}$ 90..... 41395.7
Error..... 2.293
Result..... 0.55%

Below the notch tank are shown two large cast-iron tanks, made of sectional plates 10 ft. 8 in. long by 8 ft. wide by 6 ft. 2 in. high, inside dimensions. The water from the meter could be discharged into either one of these measuring tanks by throwing over a tipple without interrupting the flow. These measuring tanks were provided with carefully constructed glass water columns with accurately graduated scales back of the glass. Baffle plates were constructed within the measuring tanks to prevent wave motion and resulting fluctuations in the water columns. Below the measuring tanks is shown the large storage supply tank 22 ft. long by 8 ft. wide by 6 ft. 2 in.

high, also built of cast-iron sectional plates, into which the upper tanks were in turn drained. The measuring tanks were very accurately calibrated by weighing on a corrected scale a unit charge of 100 lb., marking the scale through a certain range near the top and bottom of the tank for each unit charge of 100 lb.

25 The test as will be observed was run at approximately

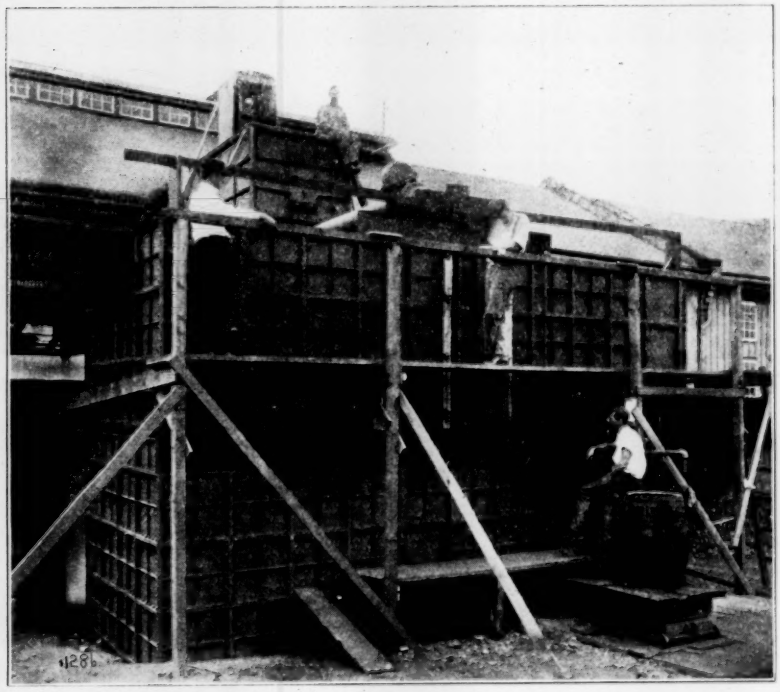


FIG. 7 ARRANGEMENT OF TESTING APPARATUS FOR PHILADELPHIA TEST

the rate of 300,000 lb. per hr. This was the maximum capacity of the recorder under test. A chart record was obtained during the test which was measured carefully in order to get the quantity for comparison with the quantity measured in the tanks below the weir tank. The recorder had a drum with a curve of flow for use at a temperature of 212 deg. As the water during the test was 73 deg. it was necessary to make a correction for this large difference in temperature. This correction has been applied in the data of the test as will be noted. Correction is also made for inaccuracy of scale as found by calibration.

Duration of test = 1:57.86 = 1.9643 hour.
By scaling chart = 1.09375 in. = 1.9643 hour.
Hence recorder clock driving chart is correct.

Then 1 in. of chart = $\frac{1.9643}{1.09375}$ = 1.796 hours.

By scaling chart (vertically) 2.5 in. = 300,000 lb. per hr.
Therefore 1 in. = 120,000 lb. per hr.
Therefore 1.796 × 120,000 = 215520 lb. = 1 sq. in. in Fig. 6C.

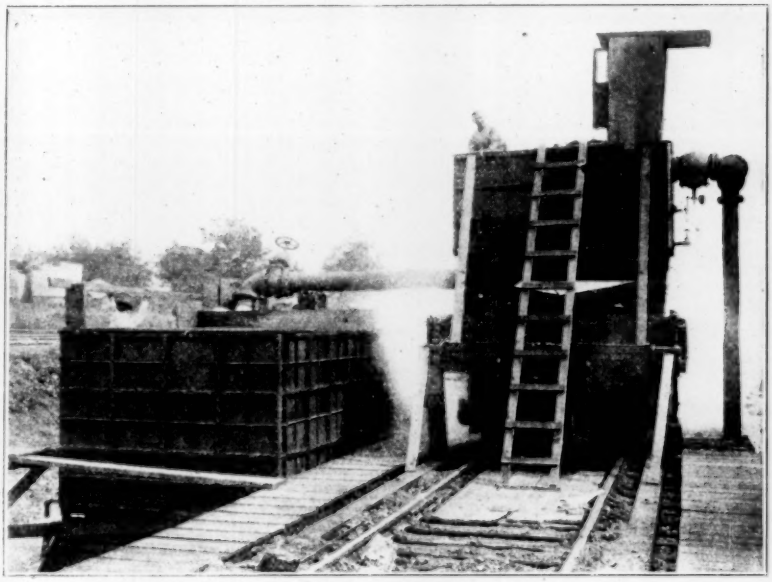


FIG. 8 ARRANGEMENT OF TESTING APPARATUS FOR PHILADELPHIA TEST

26 Integrating chart area under curve = 2.70 sq. in.
Hence 215520 × 2.7 = 581,904.0 lb. = recorder record.

27. Error in scales: each 400 lb. water as graduated at gage glass on tank is actually 400.75 lb. by correct weight, each pound of water passing through the meter cold (about 68 deg.) will increase in volume to such an extent when hot as to require the addition of 3½ per cent of the indicated chart record to the value given by the chart when the cam is cut for a range of from 175 deg. to 225 deg., average 200 deg.

Correction for scales = 1.001875
Correction for temperature = 1.035

581,904 × 1.035 =	602270.64	chart weight as corrected
595,750 × 1.001875 =	596867.03	corrected weight by tanks
Difference =	5403.61	recorder lead
Result =	0.9 per cent	recorder lead

TEST ON V-NOTCH METER NEAR NEW YORK, APRIL 1912

28 It is to be regretted that the series of tests made in April 1912 on a 400,000 lb. per hour V-notch meter cannot be described in full in this paper; the results in brief form are as follows:

29 This series of tests differs from the two tests just described

TABLE 4 TEST MADE NEAR PHILADELPHIA, PA., AUGUST 9, 1912

Consecutive Time	Elapsed Time	No. of Tank	North Bottom Reading	North Top Reading	Net North	South Bottom Reading	South Top Reading	Net South	Temperature
10:25.10*									
10:30.10	:05.00	1				26200	410	25790	
:35.07	:04.97	2	26250	100	26150				72
:40.09	:05.02	3				25875	150	25725	
:45.54	:05.45	4	26150	210	25940				72
:50.54	:05.00	5				25950	170	25780	
:55.64	:05.10	6	26150	220	25930				72
11:00.74	:05.10	7				26160	230	25930	
:05.87	:05.13	8	26330	220	26110				72
:10.96	:05.09	9				26220	210	26010	
:16.06	:05.10	10	26025	220	25805				72
:21.17	:05.11	11				25950	220	25730	
:26.29*	:05.12	12	26025	200	25825				73
:31.41	:05.12	13				26100	180	25920	
:36.53	:05.12	14	26075	200	25874				73
:41.66	:05.13	15				26030	200	25830	
:46.87	:05.21	16	26300	200	26100				
:52.00	:05.13	17				26025	160	25865	73
:57.20	:05.20	18	26000	180	25910				
12:02.39	:05.19	19				26450	160	26270	73
:07.53	:05.14	20	26000	210	25790				
:12.68	:05.15	21				26030	190	25840	73
:17.88	:05.20	22	26025	180	25845				
:22.96*	:05.08	23				26000	220	25780	74
1:57.86	:117.86		287420	2140	285280	312990	2520	210470	74
					310470				
			Total weight.....		595750				

* Marked chart and started to fill south tank.

in that the water was actually weighed on large platform scales, and hence no correction for temperature was necessary.

5-HOUR TEST ON V-NOTCH METER, APRIL 19

Water by actual weight, lb.	291965
Water by recorder, lb.	290525
Lead of scales, lb.	1440
Lead, per cent	0.49

8-HOUR TEST ON V-NOTCH METER, APRIL 20

Water by recorder, lb	267260
Water by actual weight, lb	266009
Lead by recorder, lb	1251
Lead, per cent	0.46

13-HOUR TEST ON V-NOTCH METER, APRIL 22

Water by actual weight, lb	557974
Water by recorder, lb	557775
Lead of scales, lb	199
Lead, per cent	0.03

8-HOUR TEST ON V-NOTCH METER, APRIL 23

Water by actual weight, lb	352725
Water by recorder, lb	351501
Lead of scales, lb	1224
Lead, per cent	0.34

8-HOUR TEST ON V-NOTCH METER, APRIL 24

Water by actual weight, lb	352725
Water by recorder, lb	352231
Lead of scales, lb	494
Lead, per cent	0.14

CONCLUSION

30 In conclusion it will therefore be noted that the V-notch method of measurement may be consistently depended upon within 1 per cent, even in view of fluctuating conditions of the rate of flow and also fluctuating temperatures. By using $\frac{1}{4}$ 90 degree notch weir, still greater accuracy may be obtained, and for all particularly accurate tests either $\frac{1}{2}$ 90 degree or $\frac{1}{4}$ 90 degree weir plates are recommended.

31 Attention is here called to an able discussion of the V-notch with results of tests made by James Barr, B.Sc. Carnegie Research scholar at the James Watt Engineering Laboratories, Glasgow University, 1907–1909. The synopsis of his results appear in London Engineering, April 8 and 15, 1910 editions.

DIMENSIONS OF BOILER CHIMNEYS FOR CRUDE OIL

AT SEA LEVEL AND AT VARIOUS ALTITUDES, WITH CORRECTING FACTORS FOR FUEL ECONOMIZERS

By C. R. Weymouth[1]

ABSTRACT OF PAPER

Chimney tables applicable to boiler plants using coal fuel give dimensions larger than necessary for oil fuel. It is impossible to compile a single table of correct chimney capacities applicable to different types of boilers, operating conditions, etc., for oil fuel.

This paper sets forth the elementary principles for the determination of the dimensions of chimneys, with special reference to the use of crude oil; it also presents data regarding the draft necessary for burning oil under Babcock and Wilcox boilers of various sizes, at various ratings, and a table of chimney capacities applicable to certain sizes of Babcock and Wilcox boilers.

The operating fuel economy of a crude oil plant is largely influenced by the excess air supply due to excessive draft, and the author recommends the use of lower chimneys than customary, particularly for uniform load plants, not subject to overload, as affording a means of chimney regulation of fuel economy.

Errors are shown in previously accepted formulae for variation of chimney proportions for operation at different altitudes, and correct formulae are given.

Data are given regarding the frictional resistance of fuel economizers, and a table presented showing ratio of reduction of chimney capacity, due to cooling of gases within economizer, and the resistance of economizer. Dimensions of steel chimneys are given for minimum first cost, and selection of type of chimney suitable for quick starting reserve plants is discussed.

DIMENSIONS OF BOILER CHIMNEYS FOR CRUDE OIL

AT SEA LEVEL AND AT VARIOUS ALTITUDES, WITH CORRECTING FACTORS FOR FUEL ECONOMIZERS

By C. R. Weymouth, San Francisco, Cal.

Member of the Society

The necessary heights for chimneys for fuel oil plants are much less than ordinarily supposed when the boilers are operating at rating and considerably greater than usually supposed when there are heavy overloads. A chimney of undue height causes an excessive quantity of air for combustion and permits an excessive load on the boilers, both resulting in a large waste of fuel.

2 When the chimney height is limited to that necessary for economical air supply at the desired rate of load, it will be impossible for the most careless fireman to cause a material loss of economy, either by overload or excess air supply, or abuse of boilers due to overload. In this sense the chimney may become an important and inexpensive means for boiler plant regulation and in many ways a safeguard against careless firing. Such usage is, of course, of greatest effect in plants operating at uniform load.

3 Although the writer had previously intended investigating this subject, it was not until the San Francisco earthquake of April 18, 1906, had considerably reduced the height of most masonry chimneys that he was led to an active collection of chimney data. Many of the results apparently were contradictory. Certain chimneys reduced to a height of 30 ft. gave the usual boiler capacity and others, reduced to a height of 75 ft., showed under certain conditions of service a decrease in boiler capacity.

4 If the total or potential draft of a chimney were available

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for the flow of gases through the chimney and equal to the frictional resistance within the chimney, the capacity of the chimney would then be a maximum. On the other hand, if a large percentage of the draft of a chimney were maintained at the base it is self evident that but little draft would be available to induce a flow of gases; and the greater the draft power at the base of chimney, the less the capacity of chimney in weight of gases discharged per unit of time.

5 In this respect a chimney of given diameter and height may be likened to a centrifugal fan of given diameter operating at a fixed rate of revolutions. When operating on free air discharge the capacity of the fan is very large, but when operating against an increasing pressure the rate of discharge rapidly decreases. No one would think of computing a table showing fan capacities without specifying the size of fan and the working pressure.

6 In the same sense a table of chimney capacities is incomplete which does not designate the working draft at the base of chimney; particularly for oil fuel, because of the rapid increase in draft resistance as boilers are overloaded. For an assumed set of conditions, a chimney, as with a fan, has a definite capacity, but no overload capacity.

7 Chimneys should therefore be selected, not with respect to the rated horsepower of the connected boilers, but after consideration of the maximum boiler overload and the draft resistance of boilers at overload.

8 While most chimney tables are reduced to a boiler horsepower basis, for convenience, the correct designation of capacity is the weight of gases discharged per unit of time. It follows that in determining the weight of gases from the boiler horsepower, two important factors enter: the ratio of air for combustion to that chemically required, and the efficiency of boilers at the maximum load.

9 Knowing the weight of gases to be handled, it is comparatively an easy matter to determine from existing formulæ the frictional loss due to a given chimney and flue. There are, however, but few published data indicating the draft resistance of various types of oil fired boilers under various rates of firing.

10 Practical experience has shown that altitude is an important factor affecting the capacity of boiler chimneys. Certain rules of thumb are now employed, and certain formulæ have been

incorrectly derived. The writer's treatment of this question is also given in this paper.

WEIGHT OF CHIMNEY GASES

11 In a given plant, the total weight of gases per hour to be handled by chimney is dependent upon the rated capacity of the boilers, their maximum momentary overload capacity, the necessary percentage of excess air for a given type of boiler and furnace, and the boiler efficiency for the conditions assumed.

12 Tests have indicated the efficiencies of boilers, the temperatures of waste gases, and the minimum air required for combustion. In every-day practice the negligence and ignorance of firemen, the disarrangement of furnace brickwork, the carbonization of burners and the improper adjustment of dampers make it necessary to provide for an excess of air over the minimum air supply obtainable in tests, depending on the type of boiler, the character of service, skill of firemen, etc.

13 For the best types of water-tube boilers fitted with Peabody furnace and operating at rating or at overloads, the writer has assumed as ample margin for good practice 50 per cent excess air over chemical requirements at chimney inlet. In determining draft resistance of boilers, 50 per cent excess air through the setting has been assumed for boilers at rating, and slightly less at 50 per cent overload, as explained below.

14 For poorly designed and operated furnaces, 50 per cent excess air is an insufficient allowance and the chimney proportions given in the tables must be modified by the designer to meet actual conditions.

15 When new, a boiler plant will require a minimum of air for combustion and usually there will be but little greater weight of gases passing up the chimney than into the furnaces. Allowance must be made for leaky boiler settings, long masonry flues, etc., but with steel flues air infiltration should be negligible. The writer recalls a test of an old plant having long brick flues, where the rate of flow of gases up the chimney was about double that into the connected boiler furnaces; and as a result of this leakage of cold air, the chimney temperature was very much lower than that of the gases leaving the boilers.

16 With expert firing it is possible to reduce the air supply to nearly the theoretical minimum, both at rated load and at overloads. At rating, the relatively large furnace volume makes it

possible for considerable amounts of air to pass through the furnace without entering the zone of combustion; but as the load is increased the flame more completely fills the furnace and, except with a deliberate wastage, a larger portion of the air admitted commingles with the flames and aids combustion.

17 In the computations which follow the writer assumes 14 lb. of air chemically required per lb. of oil and, based on 50 per cent excess, there would be admitted a total of 21 lb. of air per lb. of oil. To this the products of combustion and steam for atomization add slightly in excess of one pound, so that hereafter 22 lb. of chimney gases per pound of oil will be assumed.

18 To assist in the computation of the weight of chimney gases Table 1 is presented, giving calculated boiler efficiencies based on assumed conditions, as explained in the foot notes of the table.

19 The temperature of the waste gases and the gas analysis are more easily determined than the boiler efficiency and, based on these factors, the table assists in approximating the actual boiler efficiency and correspondingly the rate of oil consumption. Due allowance should be made for reduction of boiler efficiency due to any accumulation of boiler scale, sooty tubes, etc., after a period of use.

DRAFT RESISTANCE OF BOILERS

20 The draft resistance of a boiler depends on type, size, arrangement of heating surface, type and arrangement of superheater, if any, extent of baffling of gases, type of oil furnace, arrangement and area of air admission to boiler and to furnace, the excess air for combustion over chemical requirements and the rate at which the boiler is operated. The draft resistances of water tube boilers are ordinarily recorded in the rear of boiler, inside the setting, and close to the outlet damper. These drafts, therefore, do not include the resistance of damper and flue.

21 If a boiler be fired with wide open ash doors, the regulation of air being from the damper at boiler outlet, the draft resistance of the boiler will be a minimum. In boilers having a series of up and down passes, the draft within the boiler setting at the top of the first pass may entirely disappear and a positive pressure may result. This condition is due to the injector action of the burner pumping air into the furnace, and to the less

weight of the heated column of gases inside the boiler as compared with external column of air of equivalent height.

TABLE I MAXIMUM THEORETICAL EFFICIENCY FOR A GIVEN PER CENT EXCESS AIR SUPPLY AND FLUE GAS TEMPERATURE

Based on Stated Assumptions [1]

Assumed Per Cent Excess Air Supply	Assumed Temperature of Flue Gases, Deg. Fahr.										Assumed Per Cent Excess Air Supply
	375	400	425	450	475	500	550	600	700	800	
	Calculated Boiler Efficiency, Per Cent										
0	81.35	80.78	80.20	79.61	79.03	78.44	77.28	76.11	73.77	71.43	0
10	80.62	80.04	79.41	78.78	78.15	77.52	76.25	74.99	72.47	69.94	10
50	77.90	77.08	76.25	75.44	74.62	73.80	72.16	70.52	67.24	63.97	50
100	74.42	73.37	72.32	71.26	70.21	69.16	67.04	64.95	60.72	56.51	100
150	....	....	....	....	65.79	64.51	61.93	59.35	54.20	49.04	150
200	....	....	....	....	....	59.86	56.81	53.77	47.67	41.58	200
250	....	....	....	....	....	....	51.69	48.18	41.15	34.11	250
300	....	....	....	....	....	....	46.58	42.60	34.63	26.65	300

Approximate Relation between Per cent Co_2 and Excess Air Supply as per Assumptions below

Per cent Co_2	4	5	6	7	8	9	10	11	12	13	14	15	15.9
Per cent Excess Air	283	205	155	120	93.0	72.2	55.5	41.9	30.6	21.0	12.7	5.6	0

[1] Values are conditioned on the following assumptions:

Average temperature of air for combustion entering the boiler 80 deg.; humidity 80%; air per lb. of oil chemically required for complete combustion, 14 lb. b.t.u. per lb. of oil as fired, 18,500; chemical composition of the oil as follows: carbon, 86%; hydrogen, 11%; sulphur, 0.8%; nitrogen, 0.2%; oxygen, 1.0%; water, 1.0%.

Per cent of excess air stated is measured at boiler outlet and consequently includes leakage through boiler setting.

The loss by radiation has been taken as 3%. This loss varies with the size of boiler, insulation, etc., being less than indicated for larger sizes of boilers and greater for smaller sizes.

An allowance of 2% for undetermined losses has been made. This quantity is subject to considerable variation and may, in exceptionally favorable instances, be as low as 0.5%.

The following constants are taken from Marks and Davis' Steam Tables: Absolute temperature $=459.6$ deg. $+ t$ deg. Heat of vaporization at atmospheric pressure $=970.4$ b.t.u. Specific heat of superheated steam at atmospheric pressure, for the range of temperature from 212 deg. to 700 deg. fahr. $=0.47$.

This table shows conservative efficiencies primarily for use in stack calculations.

22 In boilers with brick settings there may be an escape of hot gases from the top of boiler at times, giving rise to a possible loss of economy and also to annoyance of firemen from

escaping gases. Under certain conditions it is therefore necessary to close the air admission doors partially to insure a draft at all points within the boiler setting, tending to increase the total draft resistance otherwise necessary. This effect is more noticeable at loads approximating rating, or slightly below, than at overloads, where the friction of flow of air for combustion into the boiler and furnace tends to prevent greater pressure within the setting than without.

23 The analysis of over 75 tests under various conditions, on several sizes of boilers of the same general type, fails to reveal a very close relationship between the total draft resistance and the rate of load on boiler. Other conditions being equal, it is clearly evident that the draft resistance increases slightly with an increase in size of boiler for all rates of firing. A 14-tube high Babcock and Wilcox boiler has a draft resistance greater than a boiler 9 tubes high. There is a variation of the injector action of the burner with different types, arrangement of burners, arrangement of checkerwork forming the floor of furnace, position of the burner relative to checkerwork, and the form of flame relative to checkerwork.

24 Engineers and operators vary the size and arrangement of air spaces for admission to the furnace, and the length and area of air ducts leading from the air doors to the furnace.

25 The area of air admission doors has a general relationship to boiler horsepower, but changes in area if these doors are irregular between certain sizes of boilers. In a Babcock and Wilcox boiler of 600 h.p. with a certain furnace arrangement, draft resistances at rating have been noted as low as 0.08 in. and as high as 0.2 in., this variation being due mainly to increasing quantities of air supplied for combustion. A change in furnace arrangement for economical firing gave for this same boiler a draft averaging slightly over 0.10 in. in one case and less than 0.05 in. in another case, at about the same rating and efficiency.

26 The writer's data are compiled from a series of tests made especially for the determination of boiler efficiencies and not with a view to determination of stack capacities. A series of curves was plotted and an attempt made to correct for any inequalities due to the position of air admission doors, the ratio of excess air over chemical requirements, etc. Even with these corrections, for reasons outlined above, the plotted points did not fall on smooth curves.

27 Fig. 1 shows curves of average draft resistances on Babcock and Wilcox boilers of several sizes, at various rates of firing, given for minimum observed air supply. The apparent inconsistency is due to the tests having been made by different persons

28 Fig. 2 shows curves of corrected drafts to conform as nearly as may be to a rate of air supply 50 per cent in excess of chemical requirements, the curves being rationalized with respect to each other for similar furnace settings, differences being due to variation in size of boiler, etc.

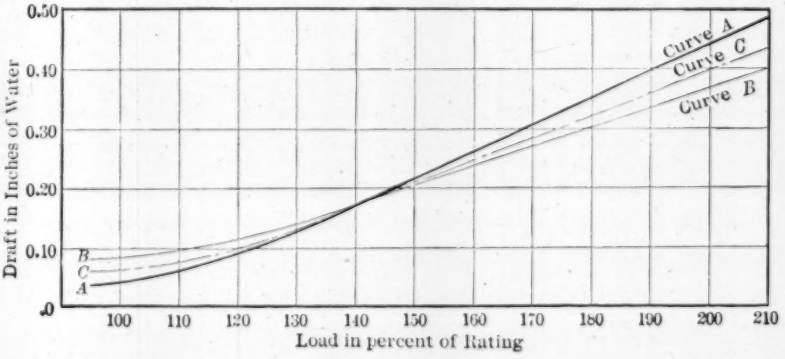

FIG. 1 DRAFT RESISTANCE OF BABCOCK AND WILCOX BOILERS WITH PEABODY FURNACES BURNING CALIFORNIA CRUDE OIL. CURVES PLOTTED THROUGH POINTS OF MINIMUM DRAFT CORRESPONDING TO MINIMUM AIR SUPPLY, TEST CONDITIONS, DRAFT MEASURED IN THIRD PASS.

Curve *A*—21 wide, 14 high, 18 ft. long, 604 h.p.[1]
Curve *B*—21 wide, 12 high, 18 ft. long, 524 h.p.
Curve *C*—14 wide, 9 high, 18 ft. long, 260 h.p.

29 All drafts in Figs. 1 and 2 are measured inside of the boiler between the end of the third pass and boiler outlet.

30 It should be noted that the draft at the rear of boiler will be somewhat dependent on the height above the floor line at the point of measurement. This height will vary from about 6 to 20 ft., depending on the type of boiler, and introduces a considerable correction for the chimney height otherwise determined. It is evident that the calculated height of chimney must be measured above the point of observed or calculated draft for the boiler in question, and not from the boiler room floor line.

31 The draft resistance between the third pass and the flue will depend upon the size, type and arrangement of boiler, area

[1]Equipped with double loop superheater.

and position of damper opening, arrangement of circulating tubes, etc.

32 The calculated draft resistance through the damper box, on the basis of velocity head, allowing ordinary coefficients for friction, does not give a result as great as the average results obtained by experiment. Draft readings are subject to error, owing to the position of draft tube as affected by velocity of

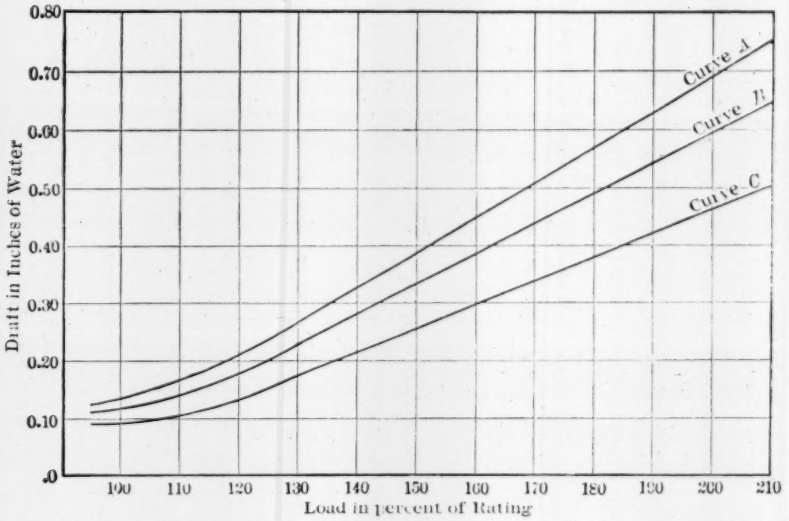

FIG. 2 DRAFT RESISTANCE OF BABCOCK AND WILCOX BOILERS WITH PEABODY FURNACES BURNING CALIFORNIA CRUDE OIL WITH DRAFT MEASURED IN THIRD PASS AND AIR SUPPLY 50 PER CENT IN EXCESS OF CHEMICAL REQUIREMENTS.

Curve *A*—21 wide, 14 high, 18 ft. long, 604 h.p.[1]
Curve *B*—21 wide, 12 high, 18 ft. long, 524 h.p.
Curve *C*—14 wide, 9 high, 18 ft. long, 260 h.p.

gases, etc. In one instance, with a Babcock and Wilcox boiler operating at 100 per cent overload, the draft observations were the same on both sides of the boiler damper, a result manifestly in error. In a 200-h.p. Babcock and Wilcox boiler working at 67 per cent overload, with economical air supply, the draft loss between the third pass and the flue side of damper with damper wide open was 0.04 in.

33 A 520-h.p. Babcock and Wilcox boiler, under good firing conditions, at slightly over rating, gave 0.03 in. loss between flue and third pass, the damper being wide open. Another test on

[1]Equipped with double loop superheater.

boilers on the same size gave 0.02 in. draft loss, at a slight overload, economical air supply.

34 A 680-h.p. Babcock and Wilcox boiler, having overhead flue, outlet between drums, gave the same draft in flue on boiler side of damper as in the third pass, a result no doubt affected by draft differences, due to difference of height of points of measurement.

35 A series of tests on a 600-h.p. Babcock and Wilcox boiler at the Redondo plant, 14 tubes high, with a double loop superheater, corrected for 50 per cent excess air supply, gave an average curve from which the following values were taken:

Percentage of Boiler Rating	Draft Resistance, Damper only, in. Water
100	0.08
125	0.11
150	0.14
175	0.16

FRICTIONAL RESISTANCE OF STEEL FLUES, ETC.

36 It is not the purpose of this paper to present new data for frictional resistances, etc., but rather to assume the correctness of established formulæ.

37 The resistance of flues will depend upon the construction, whether of steel, brick, cement, etc., and will also vary with the character of sooty coating, etc. For circular unlined steel flues after a period of use, the writer has assumed the formula

$$D = \frac{3\ W^2\ T\ L}{d^5} \qquad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

in which

D = draft loss, in. of water.
W = weight of gases flowing per second, lb.
T = absolute temperature of gases, deg. fahr.
L = length of flue, ft.
d = diameter of flue, in.

38 The coefficient 3 is approximate only and is based on information furnished by a prominent fan manufacturer, slightly increased to allow for sooty surfaces. The same source indicates that the frictional loss of brick or cement flues would be nearly double that for clean sheet iron, but the writer believes that the loss will not increase in this ratio, owing to the fact that both surfaces will eventually have a coating of soot.

39 Compared with circular flues, the resistance of those of rectangular section is greater at the same velocity by reason of the increased wetted area, being approximately 12 per cent greater for flues of square section and 15 per cent greater for flues of 1 by 1½ rectangular section than for flues of circular section.

40 The resistance of a right angle turn is taken by most authorities as equivalent to the velocity head. Reduced to the basis of the above nomenclature, we have

$$D = \frac{2.6\ W^2\ T}{d^4} \qquad [2]$$

in which

D = draft loss due to velocity head or right angle turn

41 The formulæ for flue resistance are modified for chimneys as indicated below, the difference in formulæ being, on first thought, not entirely consistent.

42 For resistance of bends of angles greater or less than 90 deg., for diverging or contracting shapes, the designer will be compelled to modify the formulæ on the basis of standard formulæ, the factor of judgment entering largely and the final results being approximate only.

43 The resistance due to a right angle turn in the breeching or at chimney has an important effect on the height of chimney, and to minimize the loss in straight runs as well as in turns, conventional practice requires an area of breeching about 20 per cent in excess of that of chimney.

DRAFT POWER OF CHIMNEY, FRICTION LOSS, ETC.

44 The exact predetermination of the potential draft of a chimney is most difficult, as it is impossible to disassociate the frictional loss within a chimney from the draft loss due to cooling, air leakage, etc.

45 It has been pointed out in Professor Gale's paper[1] that there is a decrease in the potential draft of a chimney, due to friction and cooling of gases within the chimney, to infiltration of cold air, etc., and to atmospheric currents of air in the region of the top of chimney. It has also been shown in Mr. Babcock's discussion of the paper[2] that with a quiet atmosphere there is an

[1] H. B. Gale, Theory and Design of Chimneys, Trans. Am. Soc. M. E., vol. 11, p. 451.

[2] Trans. Am. Soc. M. E., vol. 11, p. 482.

increment of draft due to the column of heated gases directly above the chimney; and further, that the form of the top of the chimney has a direct bearing on the chimney draft whenever the wind velocity is appreciable.

46 The writer would add that the deceleration of gases within the chimney, due to any cooling in the passage up the chimney, will result in a slight decrease of chimney friction on account of lessening of the volume and velocity of gases; and the conversion of the velocity head into a pressure head will also tend to overcome the downward atmospheric pressure at the top of chimney. It has been suggested to the writer that there are nodal points within the chimney owing to vibration of the chimney column and that this phenomenon may interfere with the results calculated from an elementary hypothesis.

47 The effective draft of a chimney will be reduced if ample area-ways are not provided for the inflow of air to the boiler room.

48 From the report of the Liquid Fuel Board, United States Navy, Bureau of Steam Engineering, during 1902 and 1903, the writer noted that in one particular test under forced draft conditions, where the weight of gases passing through the chimney was several times that corresponding to the natural draft conditions, giving rise to a discharge velocity of nearly 60 ft. per second, the calculated chimney friction was greater than 0.5 in. water pressure. The potential draft, due to the weight of chimney gases, was somewhat greater than 0.5 in., so that, neglecting the external influences, the draft at the base of chimney should have been practically negligible. On the other hand, the observed draft was almost as great as the potential draft, indicating a draft increment due to causes other than the difference in weight of the column of air inside and outside of chimney. On other tests the observed draft was considerably reduced, probably owing in part to the brushing aside by the wind of the hot column of gases immediately over the chimney.

49 An examination of the results of very elaborate tests made on a chimney at the Massachusetts Institute of Technology, giving draft observations at various portions, does not indicate a consistent relationship between the observed and calculated draft loss within the chimney for various points along the chimney shaft.

50 Sufficient tests have not been made to determine with accu-

racy the results from all of the influences noted, nor is it necessary that such tests be made. Chimneys must be designed with respect to the unfavorable surrounding conditions and with this idea in view it will be safe to ignore any increment in draft due to the extraordinary influences above cited.

51 Since the existing data in regard to the density of chimney gases refer particularly to coal-burning practice, it is necessary to consider possible variation with oil fuel, which is found to give a density of chimney gases somewhat greater than that with coal firing, varying with the excess air for combustion, the ultimate analysis of the crude oil, humidity of air for combustion, etc.

52 The specific volume of chimney gases is usually defined by

$$S = \frac{T}{K} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [3]$$

in which

S = cu. ft. of gases per lb.

T = absolute temperature which, according to Marks and Davis = 459.64 + temperature, fahrenheit scale.

K = a constant

For dry air, standard barometer, K = 39.78.

53 For chimney gases at a temperature of 500 deg. fahr., 50 per cent excess of dry air being supplied for combustion, K = 39.73. In this instance the value of K is 0.13 of 1 per cent greater for dry air than for chimney gases, indicating a relative lessening in the weight of chimney column to this extent.

54 For 500 deg. fahr. temperature of chimney gases, 100 per cent excess of dry air supplied for combustion, K becomes 39.74, or but slightly heavier than for chimney gases with 50 per cent excess air.

55 For atmospheric air at 80 per cent humidity, K becomes 39.29, a decrease in density as compared with dry air of 1.2 per cent.

56 For chimney gases at a temperature of 500 deg. fahr., 50 per cent excess air for combustion at 80 per cent humidity, K becomes 39.38.

57 It is thus evident that the relative density of chimney gases is slightly greater than that of atmospheric air, considering the specified 50 per cent humidity, as against the relative decrease in density when considering dry air only. It follows that with 50 per cent excess air, 500 deg. temperature of chimney gases, there

is a degree of saturation of the atmosphere less than 80 per cent humidity, at which the relative density of chimney gases is the same as for atmospheric air.

58 For 500 deg. fahr. temperature of chimney gases, 100 per cent excess air at 80 per cent humidity, the value of K becomes 39.34, approaching still more nearly than for 50 per cent excess air the relative density of the atmospheric column.

59 In these instances the greatest relative difference in the relative densities of chimney column and atmospheric air is less than ¼ of 1 per cent, which for ordinary conditions would affect the calculated draft power of the chimney by less than ½ of 1 per cent. Since these differences are small compared with the uncertain value of the correcting factor f, as will be explained, and uncertain chimney calculations in general, it may be considered safe in calculations for chimney draft to ignore the difference in relative density at chimney column and atmospheric air, assuming throughout the same relative density for both and as equivalent to the atmospheric column based on dry air. A humid atmosphere will affect the performance of a boiler plant, tending slightly towards a sluggish combustion, and by reason of a slightly increased volume and velocity of gases, there is very small increase in the frictional resistances; these effects, however, may be considered of secondary importance.

60 The calculation of the potential draft of chimney is in accordance with

$$D = fKH \qquad [4]$$

in which

f = empirical coefficient depending largely on the rate of cooling gases within the chimney

K = theoretical draft per ft. of chimney due to difference in weight of atmospheric air and chimney gases, assuming the same relative density for both as for dry air at standard barometer, the chimney column being assumed throughout at the temperature of the entering gases

H = height in ft. of chimney above point of draft measurement

For altitudes other than sea level, the draft varies directly as the normal barometer.

61 Based on the results from actual practice from a number of unlined steel chimneys, the writer has assumed the value $f =$

0.95. This factor is subject to considerable variation. In chimneys having relatively large radiating surface, and low velocity of gases, the factor f may be 0.85 or lower. If the chimney be properly designed for a certain maximum load, then, with all boilers in operation, the velocity within the chimney will be normal, and the value of f should not vary greatly.

62 A German engineer makes the factor f dependent on the difference between the mean temperature inside the chimney and the mean temperature of external air, but this formula is still at fault in neglecting the chimney velocity and cooling area of chimney surface. It does not appear possible to define the factor f by any simple rational formula.

63 The value of K is determined on the conventional basis. It is important, however, to note that in determining K, when used in connection with the factor f, the temperature should be that of the gases entering the chimney and not the mean temperature of gases within chimney.

64 Table 2 gives the calculated potential draft of a chimney 100 ft. in height for various temperatures of gases entering the chimney and for various temperatures of external air. Column A shows the theoretical draft without loss due to cooling or friction; column B the theoretical draft corrected for a temperature drop in stack and density of flue gases, based on the value $f = 0.95$; column C the net draft based on the stated friction ratio as referred to below. It is important to note that the drafts are seldom measured at the floor line, and in computing chimney heights, allowance should be made for the height above floor line, of the point of draft measurement.

65 In order to obtain the draft within chimney at the point of entering gases, it is necessary to deduct from the values given in Table 1, Column B, the frictional loss within the chimney, which may be calculated

$$D = \frac{C W^2 T H}{d^5} \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

in which the nomenclature for D, T, W and H will apply as above. C will be represented by a coefficient from practice. For steel chimneys, the writer assumes $C = 2.3$. A German authority gives a frictional coefficient for brick chimneys which reduced to the same form would give $C = 2.5$.

66 The writer has not assumed differing values for steel, brick, and concrete chimneys. While manifestly a difference will

exist, it is believed that the increased friction loss due to masonry will be somewhat offset by the greater rate of cooling of an unlined steel chimney, and with the ordinary heights required for burning oil fuel, these differences will largely disappear.

67 The "Stirling" book recommends that, for brick chimneys, the diameter be made 6 per cent greater than for steel. A German engineering association as a result of its investigations recommended for steel chimneys that the diameter be nearly one-third larger than for brick chimneys.

68 The attendant difficulties in the exact determination of draft loss are so complicated by external influences, as already pointed out, that it is probable that exact information will be long delayed.

69 Based on equation [2] for the resistance of a right angle turn and equation [1] for the resistance of a flue, a simple calculation will show that the loss due to velocity head or the loss due to a right angle turn at flue velocity is equivalent to the resistance of a length of steel flue equal to 10.4 diameters.

70 Based on equation [5] the loss due to velocity head or a right angle turn at chimney velocity will be equivalent to the resistance of a height of steel chimney equal to 13.6 diameters.

TEMPERATURE OF GASES

71 As is well known, the temperature of gases will depend upon the type of boiler, rate of firing boiler, rate of excess air supply, and general condition of boilers as affecting thorough circulation of gases and absorption of heat.

72 In economical types of boilers under test conditions, escaping temperatures are at rating about 425 deg. fahr., or slightly under, and at 50 per cent overload, 500 deg. fahr. or slightly under. With economical boilers under good firing, with tube surfaces in good condition, the boiler temperature should be about 450 deg. fahr, or under, at rating, and about 525 deg. fahr. or under, at 50 per cent overload.

73 It is necessary to allow for a drop in temperature between the boiler outlet and the chimney inlet, depending upon the length and character of the breeching, air infiltration, etc.; also upon the number of idle boilers connected to the same breeching, which may be passing cold air through dampers and which ordinarily are not close fitting.

74 In Table 5 for central chimneys, the writer has

assumed initial temperature of chimney gases 425 deg. fahr. for boilers operating at rating, and 500 deg. fahr. for boilers operating at 50 per cent overload. For chimneys directly mounted over boilers and with short breeching the relative temperature entering the chimney will be greater than indicated. Under unfavorable firing conditions and with less economical types of boilers the temperatures are materially greater than indicated.

ATMOSPHERIC TEMPERATURE

75 As is well known, but frequently overlooked, the temperature of the atmospheric column influences materially the potential draft of a chimney. By referring to Tables 2 and 3 the variation in draft, and correspondingly the variations necessary in chimney height due to range of atmospheric temperature, is appreciable. During cold weather record capacities of chimney are obtained which will not hold for extreme summer heat: the atmospheric temperature may exceed 100 deg. fahr., and in certain locations, 120 deg. fahr. The location of the boiler plant will of necessity decide the consideration given to this factor.

GENERAL FORMULÆ

76 Consideration will here be given to three types of chimneys, differing in respect to the amount of draft lost between the boiler and chimney, as follows:

77 *Case I: Independent chimney for each boiler, mounted directly on boiler, top outlet smoke connection.* In considering this case, the writer has in mind the use of Stirling boilers where an A-shaped breeching is usually employed to connect from the top flue outlet converging to the chimney diameter along lines which involve little frictional resistance within the breeching. In such cases the breeching may ordinarily be considered as the equivalent of the same height of chimney, and a separate loss due to breeching entirely ignored. Where a battery of boilers is connected to one chimney with an A-shaped overhead breeching, the case is but slightly different.

78 *Case II: Independent chimneys direct-connected to a single boiler, or a battery of two boilers, the chimney being located in the rear of boiler or boilers, a short horizontal flue being used.* In this instance the resistance of the breeching itself is almost negligible, but there is added, as compared with Case I, a greater

TABLE 2 CHIMNEY DRAFT—OIL FUEL. SHOWING THEORETICAL AND AVAILABLE DRAFT IN CHIMNEY 100 FT. IN HEIGHT ABOVE THE POINT OF DRAFT MEASUREMENT FOR VARIOUS TEMPERATURES AT SEA LEVEL

Temperature of Gases entering Chimney, Deg. Fahr.	Chimney Draft, In. of Water								
	Temperature of Outside Air = 60 Deg. Fahr.			Temperature of Outside Air = 80 Deg. Fahr.			Temperature of Outside Air = 100 Deg. Fahr.		
	A	*B*	*C*	*A*	*B*	*C*	*A*	*B*	*C*
280	0.436	0.414	0.345	0.382	0.363	0.303	0.332	0.315	0.263
300	0.464	0.441	0.367	0.410	0.390	0.325	0.359	0.341	0.284
320	0.490	0.466	0.388	0.435	0.413	0.344	0.385	0.366	0.305
340	0.514	0.488	0.407	0.460	0.437	0.364	0.409	0.389	0.324
360	0.537	0.511	0.425	0.483	0.459	0.383	0.432	0.411	0.342
380	0.560	0.532	0.444	0.505	0.480	0.400	0.455	0.432	0.360
400	0.581	0.550	0.460	0.527	0.501	0.417	0.476	0.452	0.377
425	0.606	0.576	0.480	0.552	0.524	0.437	0.501	0.476	0.397
450	0.630	0.598	0.499	0.576	0.547	0.456	0.525	0.498	0.416
475	0.652	0.619	0.516	0.598	0.568	0.475	0.547	0.520	0.433
500	0.674	0.640	0.534	0.619	0.588	0.490	0.569	0.541	0.451
525	0.694	0.659	0.550	0.639	0.607	0.506	0.589	0.560	0.466
550	0.713	0.678	0.565	0.659	0.627	0.522	0.608	0.576	0.482
575	0.731	0.694	0.579	0.677	0.643	0.536	0.626	0.595	0.496
600	0.749	0.712	0.593	0.694	0.657	0.550	0.644	0.612	0.510
625	0.765	0.727	0.606	0.711	0.675	0.563	0.660	0.627	0.523
650	0.781	0.742	0.619	0.727	0.691	0.576	0.676	0.642	0.535
675	0.796	0.756	0.630	0.742	0.705	0.588	0.691	0.656	0.547
700	0.811	0.768	0.642	0.756	0.718	0.599	0.706	0.671	0.559
725	0.825	0.783	0.653	0.770	0.731	0.610	0.720	0.684	0.570
750	0.838	0.797	0.664	0.783	0.744	0.620	0.733	0.697	0.581

Column A shows the theoretical draft due to a height of chimney of 100 ft. and $=KH$ where:

H = height in feet of chimney above point of draft measurement = 100.

K = theoretical draft per foot of chimney due to difference in weight of atmospheric air and chimney gases at stated temperatures, assuming same relative density for both as for dry air = one one-hundredth of value given in column *A*.

Column B shows draft in *A* corrected for temperature drop in stack and density of flue gases. $B = f\,A = f\,KH.f$ is variable, but assumed above as 0.95.

Column C shows net draft, being draft in column *B* corrected for friction, assumed as $\frac{3}{4}$ of draft in column *B*.

Chimney friction should be $\frac{1}{6}$ total draft for minimum first cost of chimney when mounted directly over boiler or battery of two boilers. This does not apply to horizontal breeching or to central chimneys combining several losses.

For altitude draft varies directly as normal barometer.

TABLE 3 APPROXIMATE MAXIMUM CAPACITY; ACTUAL BOILER HORSEPOWER; STEEL CHIMNEYS; SIZES FOR MINIMUM COST OF CHIMNEY; INDEPENDENT TYPE DIRECTLY MOUNTED OVER SINGLE OR DOUBLE BOILER SETTING WITH TOP OUTLET; 50 PER CENT EXCESS AIR SUPPLY; SEA LEVEL. OIL FUEL.

Temp. of Gases entering Chimney, Deg. Fahr.		300	400	500	600	700	Net Draft required at Point of Measurement In. of Water	Height of Chimney in Ft. above Point of Draft Measurement, for Various Temperatures of Entering Gases and Outside Air, corresponding to Tabulated Draft														
Assumed Boiler Efficiency		80.4	77.1	73.8	70.5	67.3		300 Deg.			400 Deg.			500 Deg.			600 Deg.			700 Deg.		
Lb. of Chimney Gases per B.h.p.		49.5	51.6	54.0	56.3	59.0		Temperature of Air, Deg. Fahr.													
Diam., In.	Area, Sq. Ft.	Actual Boiler Horsepower Temp. Outside Air, 80 Deg. Fahr.						60	80	100	60	80	100	60	80	100	60	80	100	60	80	100
18	1.77	61	62	61	59	56	0.01	2.7	3.1	3.5	2.2	2.4	2.7	1.9	2.0	2.2	1.7	1.8	2.0	1.6	1.7	1.8
21	2.41	90	92	90	87	82	0.02	5.5	6.2	7.0	4.3	4.8	5.3	3.7	4.1	4.4	3.4	3.7	3.9	3.1	3.3	3.6
24	3.14	125	127	125	121	115	0.03	8.2	9.2	10.6	6.5	7.2	8.0	5.6	6.1	6.7	5.1	5.5	5.9	4.7	5.0	5.4
27	3.98	168	172	168	163	154	0.04	10.9	12.3	14.1	8.7	9.6	10.6	7.5	8.2	8.9	6.8	7.3	7.8	6.2	6.7	7.2
30	4.91	219	223	219	212	201	0.05	13.7	15.4	17.6	10.9	12.0	13.3	9.4	10.2	11.1	8.5	9.2	9.8	7.8	8.4	9.0
33	5.94	278	283	278	268	255	0.06	16.3	18.5	21.1	13.0	14.4	15.9	11.2	12.2	13.3	10.1	11.0	11.8	9.4	10.0	10.7
36	7.07	345	352	345	334	317	0.07	19.1	21.6	24.7	15.2	16.8	18.6	13.1	14.3	15.5	11.8	12.8	13.7	10.9	11.7	12.5
39	8.30	422	430	422	408	387	0.08	21.8	24.6	28.1	17.4	19.2	21.2	15.0	16.3	17.8	13.5	14.6	15.7	12.5	13.4	14.3
42	9.62	508	518	508	490	465	0.09	24.6	27.7	31.7	19.5	21.6	23.4	16.8	18.4	20.0	15.2	16.5	17.6	14.0	15.0	16.1
45	11.05	602	615	602	583	553	0.10	27.3	30.8	35.2	21.7	24.0	26.5	18.7	20.4	22.2	16.9	18.3	19.6	15.6	16.7	17.9
48	12.57	708	723	708	685	650	0.20	54.5	61.6	70.4	43.4	48.0	53.0	37.4	40.8	44.4	33.8	36.6	39.2	31.2	33.4	35.8
54	15.90	951	971	951	919	872	0.30	81.9	92.4	106	65.1	72.0	79.5	56.1	61.2	66.6	50.7	54.9	58.8	46.8	50.1	53.7
60	19.64	1240	1260	1240	1197	1134	0.40	109	123	141	86.8	96.0	106	74.8	81.6	88.8	67.6	73.2	78.4	62.4	66.8	71.6
66	23.76	1570	1600	1570	1520	1440	0.50	137	154	176	109	120	133	93.5	102	111	84.5	91.5	98.0	78.0	83.5	89.5
72	28.27	1950	1990	1950	1890	1790	0.60	163	185	211	130	144	159	112	122	133	101	110	118	93.6	100	107

TABLE 3—*Continued*

78	33.18	2390	2430	2390	2310	2190	0.70	191	216	247	152	168	186	131	143	155	118	128	137	109	117	125
84	38.49	2870	2930	2870	2770	2630	0.80	218	246	281	174	192	212	150	163	178	135	146	157	125	134	143
90	44.18	3410	3480	3410	3300	3130	0.90	246	277	317	195	216	234	168	184	200	152	165	176	140	150	161
96	50.27	4010	4090	4010	3870	3670	1.00	273	308	352	217	240	265	187	204	222	169	183	196	156	167	179

Conversion Factors for 60 Deg. and 100 Deg. Atmosphere

Temp. of Gases	300	400	500	600	700
Air at 60 Deg.	1.06	1.05	1.04	1.04	1.04
Air at 100 Deg.	0.935	0.948	0.960	0.965	0.970

This table indicates diameter of chimney for minimum first cost of chimney when cost is proportional to the product of diameter and height. While capacities apply to any arrangement of chimney, cost is minimum only when gases flow vertically into chimney without horizontal breeching and without loss due to right angle turn of gases into chimney. Diameter for minimum cost is independent of draft resistance, if height be selected from the table corresponding to total draft resistance of boiler and damper outlet. For total chimney height above floor line, add to stated heights the actual height above floor line to point of draft measurement. For stated condition of minimum chimney cost, draft resistance is equal to one-sixth of static draft and this allowance has been made in table. See Table 1 for basis of assumed boiler efficiencies, and Table 2 for basis of draft calculations. If boilers have superheaters increase b.h.p. to include heat absorbed by superheater.

draft loss in the damper outlet and turn into flue, also the loss due to a right angle turn of the gases on entering chimney.

79 *Case III: A central chimney connected to a battery of two or more boilers by means of a common flue of appreciable length.* In the computation for this case, the writer will assume the prevailing practice in Babcock and Wilcox installations, the damper and flue outlet being directly in the rear and there being a straight run of flue to chimney. In this instance there is involved not only the loss through the damper outlet, but the loss due to the turn from the damper outlet into the breeching, loss due to friction of the breeching, and also that due to the right angle turn from the breeching into the chimney. A modification of this arrangement would include installations where there is a resistance due to turns of the breeching in which instances losses may be minimized by avoiding sharp bends.

80 The fundamental formula, ignoring economizers for the present, is written by equating the potential draft of chimney and the sum of various resistances

$$fKH = D_1 + D_2 + D_3 + D_4 + D_5 + D_6 + D_7 \ldots\ldots [6]$$

in which

f, K, H are as heretofore stated

D_1 = draft resistance of boiler measured on boiler side of damper

D_2 = draft resistance due to damper box

D_3 = draft resistance due to right angle turn into breeching

D_4 = draft resistance due to straight length of breeching

D_5 = draft resistance due to turns in breeching

D_6 = — draft resistance due to turn from breeching into chimney

D_7 = draft resistance of chimney calculated in accordance with [5]

81 D_1 and D_2 usually depend on the boiler in question, and required load or overload. Since the flue area is usually fixed by the chimney area, the values of D_3, D_4, D_5, D_6 and D_7 are interrelated as more fully explained below.

82 When investigating an existing plant under assumed load conditions, it is merely necessary to calculate the various load resistances, comparing their total with that which the chimney is capable of producing.

83 For Case I, D_3, D_4, D_5 and D_6 disappear, and the equation becomes

$$fKH = D_1 + D_2 + \frac{2.3\ W^2 HT}{d^5} \quad \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

from which

$$\text{B.h.p.} = \frac{60^2}{G} \sqrt{d^5} \sqrt{\frac{fKH - (D_1 + D_2)}{2.3\ HT}} \quad \ldots\ldots\ldots\ldots [8]$$

in which

G = lb. of chimney gases per b.h.p. per hour

84 Table 3 gives chimney capacities for Case I based on selection of height and diameter for minimum first cost of chimney as explained below.

85 The treatment for Case II is simple, after considering the more general Case III. For the latter it will be assumed that all flues are rectangular in shape, having a depth of 1.5 times the width, giving a flue resistance 15 per cent in excess of that for a circular flue of equal area and length, which length of flue is ten times the chimney diameter; that flue area is maintained substantially its entire length; and that owing to the decreasing velocities at increasing distances from chimney, the actual resistance of the straight run of breeching will be practically equivalent to that having uniform velocity its entire length, but 5 chimney diameters in length instead of 10 chimney diameters its actual length. D_1 will be as found from test. D_2 and D_3 cannot easily be separated and will be considered as D_2 properly evaluated from test. Then

$$D_4 = \frac{2.3\ W^2 T \left(\frac{5d}{12}\right)}{d^5} \quad \ldots\ldots\ldots\ldots\ldots\ldots [9]$$

D_5 disappears from Case III; but when breeching contains a right angle turn, D_5 may be taken as equivalent to the resistance of chimney height of 9.4 chimney diameters. Then

$$D_6 = \frac{2.3\ W^2 T \left(\frac{13.6d}{12}\right)}{d^5} \quad \ldots\ldots\ldots\ldots\ldots\ldots [10]$$

Equation [6] may now be written

$$fKH = (D_1 + D_2) + \frac{2.3\ W^2 T}{d^5} \left(H + \frac{18.6d}{12}\right) \quad \ldots\ldots\ldots [11]$$

The coefficient $\frac{18.6}{12} = 1.55$; in the table for central chimneys this term, for simplicity of calculation, has been taken as 1.5.

86 As equation [11] involves terms of the fourth and fifth powers, it is simpler to calculate the boiler horsepower of chimney from assumed diameter and height, when [11] finally becomes

$$\text{B.h.p.} = \frac{60^2}{G} \sqrt{d^5} \sqrt{\frac{fKH - (D_1 + D_2)}{2.3\ T\ (H + 1.5d)}} \quad \ldots\ldots\ldots\ldots [12]$$

87 In computing the table of capacity for central chimneys, Case III, $D_1 + D_2$ has been taken as 0.15 in. for boilers at rating and 0.3 in. for boilers for 50 per cent overload, which values are conservative for standard boilers, 12 tubes high; for 9 high boilers the height may be slightly reduced, and for 14 high boilers the height should be increased in keeping with draft resistance data above given. Table 4 gives calculated capacities for Case III based on various other assumed data given in tables and explained in the footnotes.

88 For Case II, in which the resistance of straight run of breeching is neglected, the coefficient of d, 1.5, in the term $(H + 1.5\ d)$, equation [12] becomes instead, $\frac{13.6}{12}$ or 1.13; where one chimney is connected to two boilers symmetrically arranged there is a right angle turn in breeching, as well as into chimney, and the above coefficient becomes $\frac{13.6 + 9.4}{12}$ or 1.92 approximately.

89 For Case III, considering one right angle turn in breeching, the coefficient of d becomes 2.33.

90 In the foregoing it is assumed, as stated above, that the total flue area is 20 per cent greater than chimney area.

91 A study of the term $(H + 1.5\ d)$, equation [12], shows that for the conditions assumed, the resistance of flue and turn into chimney, both dependent on chimney diameter, is of as much influence on chimney performance as the resistance of chimney itself, whenever the height of chimney above the point of draft measurement is less than 12×1.5 or 18 diameters. Thus, in a 5-ft. chimney the above-named friction losses equal the chimney friction when chimney height above point of draft measurement is 90 ft. or less.

CHIMNEY PROPORTIONS FOR MINIMUM FIRST COST OF CHIMNEY

92 It is obvious that an indefinite increase in the diameter of chimney will not decrease the height below that necessary to maintain the draft corresponding to the fixed boiler resistances. An increase in height will reduce the diameter only to the point where the added draft per foot of height equals the chimney resistance per foot of height. Between these two extremes there are numberless combinations of height and diameter which will maintain the desired draft at base of chimney or the rear of boiler. But for a given set of conditions and for the type of construction under consideration there is but one height and one diameter resulting in a chimney of least first cost.

93 The simplest case is that for steel chimneys having a given average thickness of material, the cost of chimney being proportional to the product of height and chimney diameter.

94 For Case I, the final equation for minimum first cost, derived by the method of differential calculus, is reducible to simplest form. A chimney of least first cost, for the simple case stated, for a constant maintained draft inside chimney at chimney base, is that in which the frictional resistance of chimney is one-sixth of the potential draft of chimney.

95 This relationship was first stated in the book "Stirling," but incorrectly, as regards central chimneys as well as direct mounted chimneys, as in Case I. For Cases II and III this relationship will not apply, and no simple formula can be derived for these cases, owing to the resistance of breeching and chimney inlet, the latter involving terms of the fourth power. Resistance of chimney and breeching introduces terms of the fifth power, the final equation of minimum cost being too complex for ordinary engineering purposes. Equating the chimney resistance and one-sixth of potential draft of chimney we have

$$\frac{1}{6} fKH = \frac{2.3\ W^2\ TH}{d^5} \qquad [13]$$

It is apparent that the term H disappears, simplifying [13] to

$$\text{B.h.p.} = \frac{60^2}{G} \sqrt{d^5} \sqrt{\frac{\frac{1}{6} fK}{2.3\ T}} \qquad [14]$$

96 The draft at base of chimney becomes, by the above relationship, five-sixths of potential draft of chimney. Column C, Table 1, gives chimney drafts on the above basis. Table 3 gives

TABLE 4.—PART I APPROXIMATE MAXIMUM CAPACITY; ACTUAL BOILER HORSE-POWER; STEEL CHIMNEYS WITH SHORT FLUES; CHIMNEYS CENTRALLY LOCATED; STATIONARY B. & W. BOILERS.

BASED ON ATMOSPHERIC TEMPERATURE OF 80 DEG. FAHR.; SEA LEVEL, AND OTHER ASSUMED DATA. OIL FUEL

Ratio of actual boiler h.p. to rated boiler h.p., per cent	100
Draft in inches of water at boiler outlet, chimney side of damper	0.15
Corresponding excess air through boiler, per cent	50
Assumed excess air supply for determining boiler efficiency, chimney diameter and draft resistance of chimney and breeching, per cent	50
Assumed temperature of gases leaving boiler, deg. fahr.	450
Assumed temperature of gases entering chimney, deg. fahr.	425
Assumed boiler efficiency, working, not test conditions, per cent	75.4
Assumed lb. of chimney gases per actual boiler h.p.	52.8

Diameter, In.	Area, Sq. Ft.	Height in Feet above Boiler Room Floor Line											
		50	60	70	80	90	100	110	120	130	140	150	160
18	1.77	57	77	88	97	103	108	112	115	118	120	122	124
21	2.41	81	109	127	139	149	156	162	167	171	175	178	180
24	3.14	110	149	173	190	204	214	222	229	235	240	245	249
27	3.98	144	194	226	250	268	282	293	303	311	318	325	330
30	4.91	182	247	288	318	342	361	376	388	400	409	417	424
33	5.94	224	306	358	396	427	450	470	486	500	513	523	533
36	7.07	272	372	436	483	521	550	575	596	614	629	642	655
39	8.30	325	444	521	582	626	662	692	718	740	758	776	790
42	9.62	382	522	615	686	741	785	821	852	880	902	923	940
45	11.05	444	610	719	801	869	920	961	1000	1030	1060	1080	1110
48	12.57	512	702	830	927	1000	1060	1120	1160	1200	1230	1260	1290
54	15.90	660	911	1080	1210	1310	1390	1460	1520	1570	1610	1660	1700
60	19.64	827	1150	1360	1520	1660	1760	1860	1940	2000	2060	2110	2160
66	23.76	1020	1410	1670	1880	2050	2180	2300	2400	2480	2560	2630	2690
72	28.27	1220	1700	2020	2280	2490	2650	2790	2910	3030	3130	3210	3290
78	33.18	1450	2020	2400	2710	2970	3170	3340	3490	3620	3750	3850	3950
84	38.49	1690	2370	2830	3190	3490	3730	3940	4130	4280	4430	4560	4670
90	44.18	1960	2740	3280	3710	4060	4350	4600	4810	5000	5170	5320	5460
96	50.27	2250	3150	3770	4260	4670	5000	5300	5560	5780	5980	6160	6320
102	56.75	2560	3580	4290	4850	5320	5710	6050	6340	6610	6850	7060	7250
108	63.62	2880	4030	4840	5480	6030	6470	6850	7180	7500	7770	8010	8230
114	70.88	3220	4510	5440	6170	6780	7280	7720	8100	8440	8750	9050	9300
120	78.54	3580	5030	6060	6870	7560	8130	8630	9060	9440	9800	10100	10400
126	86.59	3960	5580	6710	7630	8380	9030	9590	10100	10510	10900	11300	11600
132	95.03	4390	6150	7420	8420	9260	9990	10600	11200	11700	12100	12500	12900
138	103.9	4800	6730	8150	9270	10200	11000	11700	12300	12800	13300	13800	14200
144	113.1	5230	7370	8920	10200	11200	12000	12800	13500	14100	14600	15100	15600
156	132.7	6210	8730	10600	12000	13200	14300	15200	16000	16700	17400	18000	18600
168	153.9	7190	10200	12300	14000	15500	16800	17800	18800	19700	20500	21200	21900
180	176.7	8350	11800	14300	16200	17900	19400	20700	21800	22900	23800	24700	25400

Effective height of chimney or height above point of draft measurement assumed 12 ft. less than tabulated heights corresponding to 12 high B. & W. boilers, 7 ft. furnace height. Draft resistance of boiler correct for 12 high boilers; capacities somewhat less for 14 high and greater for 9 high. Capacity reduced by long breeching. For correction for altitude, Table 5; or draft calculations, Table 2; for boiler efficiencies, Table 1; for capacity independent chimneys, able 3. When building conditions permit, select chimneys of least height in table for minimum

TABLE 4—PART 2 APPROXIMATE MAXIMUM CAPACITY; ACTUAL BOILER HORSE-POWER; STEEL CHIMNEYS WITH SHORT FLUES; CHIMNEYS CENTRALLY LOCATED; STATIONARY B. & W. BOILERS

BASED ON ATMOSPHERIC TEMPERATURE OF 80 DEG. FAHR.; SEA LEVEL, AND OTHER ASSUMED DATA. OIL FUEL

Ratio of actual boiler h.p. to rated boiler h.p., per cent 150
Draft in inches of water at boiler outlet, chimney side of damper 0.30
Corresponding excess air through boiler, per cent ..... Less than 50
Assumed excess air supply for determining boiler efficiency, chimney diameter and draft resistance of chimney and breeching, per cent 50
Assumed temperature of gases leaving boiler, deg. fahr. 525
Assumed temperature of gases entering chimney, deg. fahr. 500
Assumed boiler efficiency, working, not test conditions, per cent 73.0
Assumed lb. of chimney gases per actual boiler h.p. 54.6

Diameter, In.	Area, Sq. Ft.	Height in Feet above Boiler Room Floor Line								
		80	90	100	110	120	130	140	150	160
18	1.77	63	75	84	91	96	101	104	108	110
21	2.41	90	108	121	131	139	146	151	156	160
24	3.14	123	148	166	180	191	201	208	215	221
27	3.98	161	195	219	238	253	265	276	285	293
30	4.91	206	249	280	304	324	340	354	366	377
33	5.94	256	310	349	381	405	426	444	459	472
36	7.07	312	379	427	466	497	523	545	564	581
39	8.30	376	455	514	561	599	631	657	681	701
42	9.62	443	539	609	665	711	749	782	810	835
45	11.05	518	630	713	779	834	879	918	952	981
48	12.57	599	729	827	904	967	1020	1070	1110	1140
54	15.90	779	951	1080	1180	1270	1340	1400	1460	1500
60	19.64	985	1200	1370	1500	1610	1710	1790	1860	1920
66	23.76	1220	1490	1700	1860	2000	2120	2220	2310	2390
72	28.27	1470	1810	2060	2260	2430	2580	2710	2820	2910
78	33.18	1750	2150	2460	2710	2910	3090	3250	3380	3500
84	38.49	2060	2530	2900	3190	3440	3650	3840	4000	4150
90	44.18	2390	2950	3370	3720	4010	4260	4480	4670	4850
96	50.27	2750	3390	3880	4290	4630	4920	5180	5400	5610
102	56.75	3140	3870	4440	4900	5290	5630	5930	6190	6430
108	63.62	3550	4380	5020	5550	6000	6390	6730	7030	7300
114	70.88	3990	4920	5650	6250	6760	7200	7590	7930	8250
120	78.54	4440	5490	6310	6990	7560	8060	8490	8890	9240
126	86.59	4930	6100	7020	7770	8410	8970	9460	9900	10300
132	95.03	5450	6740	7760	8600	9310	9930	10500	11000	11400
138	103.9	5990	7420	8530	9460	10300	10900	11600	12000	12600
144	113.1	6550	8120	9350	10400	11200	12000	12700	13300	13800
156	132.7	7760	9630	11100	12300	13400	14300	15100	15800	16500
168	153.9	9060	11300	13000	14400	15700	16800	17700	18600	19400
180	176.7	10500	13000	15100	16700	18200	19500	20600	21600	22600

cost of chimney. Maximum chimney capacities stated are for uniform continuous load equally divided all boilers. For large plants subject to hand regulation, or for swinging load, reduce capacity from 10 to 20 per cent. If boilers have superheaters increase b.h.p. to include heat absorbed by superheater. Breeching 20 per cent excess stack area, without turns, length not exceeding 10 chimney diameters; boilers uniformly distributed along breeching; breeching depth = 1.5 × width. For capacities of chimneys with economizers, see Table 7. For economizer draft resistance, see Table 6.

capacities of economical chimneys for various assumed conditions, also heights for various required drafts. It will be noted that diameter is dependent only on boiler horsepower, and height dependent only on draft resistance of the boiler in question, information which must be supplied by the designer. Table 3 indicates chimneys of relatively large diameter but moderate height. Whenever conditions compel greater chimney height than given in table, the diameter may be reduced to maintain the required draft at chimney base, in accordance with equation [7].

97 The low chimneys indicated by Table 3 are noteworthy. For example: Assuming a Stirling boiler, requiring 0.1 in. draft at rating, with 450 deg. temperature, draft measurement 15 ft. above floor line, for 60 deg. atmosphere the height of chimney by interpolation is about 20 ft., to which must be added the 15 ft., making a total height of 35 ft. above floor. Under test conditions the draft in certain sizes is as low as 0.05 in., for which the height would be 11 ft. plus 15 ft. or 26 ft. above floor. At 100 per cent overload this total height would be increased to from 50 to 100 ft., depending on assumed draft resistance of boiler, atmospheric temperature, etc.

98 In general, it may be said that for Cases II and III the chimney of least first cost will be of moderate height, although this height will of necessity be somewhat greater than indicated by Table 3 for Case I. The least height in Table 4 corresponds to minimum first cost for the range of height intervals chosen. A cut-and-try method must be resorted to for greater refinement.

REDUCTION OF STACK CAPACITY AT ALTITUDES

99 The writer is not aware of any treatise dealing correctly with stack capacities at altitudes, either for coal or for oil fuel. Heretofore it has been assumed that the stack height increases inversely as the barometric pressure and that the stack diameter increases inversely as the two-fifths power of the barometric pressure, all other boiler and chimney data remaining constant. This relationship is based on the assumption of constant draft, measured in inches of water at the base of chimney for a given rate of firing of boilers, either expressed in lb. of coal per sq. ft. of grate surface or in ratio of rated capacity when burning fuel oil, regardless of altitude, an assumption wholly incorrect.

100 Whenever a chimney is working at maximum capacity

either at sea level or at an altitude, it is evident that the entire potential draft is utilized in overcoming the various resistances, each proportional to the square of the velocity, which varies in different points in the path of the gases.

101 Owing to the fixed areas along the path, all velocities may be related to a common velocity, say that within the chimney, and therefore all resistances in a given boiler plant for a given temperature of gases, etc., may be expressed as proportional to the square of the chimney velocity.

102 The total resistance to flow in terms of velocity head may be expressed in terms of the weight of a column of external air, and the numerical value of this head will be independent of the barometric pressure. Likewise the potential draft of a chimney, expressed in height of column of external air, will be numerically independent of the barometric pressure.

103 It is therefore evident that if a given boiler plant with connected chimney, operated with a fixed fuel, be transplanted from sea level to an altitude and if the temperatures remain constant (an assumption not entirely correct), the total draft head, measured in height of column of external air, will be numerically constant. The velocity of chimney gases will therefore remain the same at altitude as at sea level, and the weight of gases flowing per second with fixed velocity will be proportional to the atmospheric density, or inversely proportional to the normal barometric pressure.

104 Hence, except for errors due to assumption of constant temperatures of gases, it may be stated that the maximum capacity of a given boiler plant with a fixed size of chimney and fixed fuel will decrease at altitudes higher than sea level at a rate directly proportional to barometric height. For example: At an altitude of approximately 5764 ft., the normal barometric pressure would be 80 per cent of the barometer at sea level, and a chimney connected to a given plant, giving a maximum capacity of 1000 b.h.p. at sea level, would give not more than 800 b.h.p. at this altitude. However, corresponding to the reduction in rate of load on boilers, there would be a decrease in the temperature of boiler and chimney gases, tending under normal conditions to a still further reduction in maximum capacity.

105 An error commonly made in the consideration of stack capacities at altitudes is to assume that a given grade of fuel at a fixed boiler rating will require at an altitude the same draft,

measured in inches of water, at boiler damper as at sea level. It is evident that to develop a given boiler horsepower requires a constant weight of chimney gases and air for combustion. Hence as the altitude is increased the density is decreased and, correspondingly, the velocity through the furnace, the bed of coal, or the firebrick checkerwork and the boiler passes, must be greater at altitude than at sea level. The mean velocity, therefore, for a given boiler horsepower and constant weight of gases will be inversely proportional to the barometric pressure and the velocity head, measured in column of external air, will be inversely proportional to the square of the barometric pressure.

106 It is possible and usually desirable in coal burning to select a differently arranged boiler for altitudes, particularly in the West, increasing the relative grate surface not only because of reduction of draft loss, but also by reason of the poorer grades of coal, excessive ash content, etc. These considerations, however, do not apply with oil fuel, although it is possible but not desirable to modify the arrangement of the boiler, reducing somewhat its draft resistance.

107 A given chimney with gases at a fixed velocity, temperature, etc., will result in a draft head, measured in height of column of external air, numerically independent of the barometer. The absolute draft pressure, measured in inches of water, will therefore be inversely proportional to barometric pressure.

108 It follows that for chimneys operating at altitude, it is necessary not only to increase the height, but also the chimney diameter. Corresponding to the increase in height, there is an added resistance within the chimney owing to the friction of the additional height. This frictional loss can be compensated by a suitable increase in the diameter, and when so compensated it is evident that the chimney height must be increased at a ratio inversely proportional to the square of the normal barometric pressure.

109 For Case I in which the chimney is mounted directly on top of the boiler, the gases entering the chimney without right angle turn, a simple calculation, based on equation [8], will indicate that the diameter of the chimney must be increased inversely as the two-fifths power of the barometric column. For example, for a height of 5764 ft. the normal barometer would be 80 per cent of that at sea level, and the height increases in the ratio of

1.5625 instead of 1.25 as is commonly assumed; the ratio of diameters will be 1.09.

110 If a chimney having height and diameter for minimum first cost be selected for sea level corresponding to a certain rating of boilers, then for Case I the modification indicated will still result in the most economical chimney for operation at altitude.

111 For Cases II and III the effective chimney height may still be increased in inverse proportion to the square of the normal barometer, and the diameter of chimney and breeching must also be increased in such ratio that the modified chimney and breeching will have the same frictional resistance at sea level as before increasing the chimney height at sea level. In other words, the coefficient of resistance of chimney and breeching must remain a constant. In the general Case III it was necessary to assume a relationship between flue length and chimney diameter; however, with the same boiler plant transplanted to an altitude, there is not, of necessity, any change in flue length. Therefore, if length and diameter of flue remain unchanged at altitude its coefficient of resistance remains constant.

112 If the chimney diameter be increased at altitude in the ratio set forth for Case I and the breeching be increased to 20 per cent excess area, as for sea level, then with constant length of breeching the coefficient of resistance of both breeching and chimney inlet actually decreases, indicating that by applying to Cases II and III the same correcting factor as for Case I, the modified chimney will not only be ample, but slightly larger than necessary.

113 The general formulæ for altitudes are derived as follows: Let

P_o = normal barometric pressure at sea level
P_a = normal barometric pressure at altitude, and

$$R = \frac{P_o}{P_a}$$

Then if K, D_1, D_2, etc., still indicate values for sea level, and H and d new values at altitude, equation [7] becomes

$$\frac{fKH}{R} = R\left(D_1 + D_2 + \frac{2.3\ W^2TH}{d^5}\right) \quad \ldots\ldots\ldots\ldots [15]$$

also equation [8] becomes

$$\text{B.h.p.} = \frac{60^2}{G}\sqrt{d^5}\,\frac{\sqrt{\frac{fKH}{R^2}-(D_1+D_2)}}{\sqrt{2.3\,HT}} \quad \ldots\ldots\ldots\ldots [16]$$

Likewise equation [12] becomes

$$\text{B.h.p.} = \frac{60^2}{G}\sqrt{d^5}\,\frac{\sqrt{\frac{fKH}{R^2}-(D_1+D_2)}}{\sqrt{2.3T\,(H+1.5d)}} \quad \ldots\ldots\ldots\ldots [17]$$

114 Equation [17] assumes breeching length 10 chimney diameters, which is not necessarily the case, as pointed out above.

115 It is apparent that T is not modified by altitude, with the same rate of load on boilers and the same excess air.

116 Table 5 gives correcting factors for various altitudes.

MODIFICATION OF CHIMNEY CAPACITY DUE TO FUEL ECONOMIZERS

117 Owing to the comparatively small draft required when burning crude oil, it is unnecessary to resort to mechanical draft when installing fuel economizers.

118 It is self-evident that if economizers be added to an existing boiler plant, the available draft at boiler outlet will be decreased, not only by reason of the added frictional resistance of fuel economizers, but also because of the lowering of the temperature of gases within the chimney. It is also clear that the chimney dimensions must be modified to counteract these influences.

119 For the purpose of this paper, the writer has assumed the economizer dimensions as given in the Green Fuel Economizer Company's catalogue. In determining the draft resistance of economizers there is involved not only the possible increased number of flue turns due to the installation of economizer, but also the frictional resistance of economizer tubes.

120 The writer has assumed that an economizer should be installed with an ample area for the flow of gases with the economizer "side dampers" closed, and the resistance of the economizers has been determined on this basis; the opening of side dampers then provides latitude for increased draft at boiler.

121 The resistance data for economizers indicates a lack of concordance, doubtless due to the fact that draft observations are usually made incidental to boiler and economizer tests and not for the specific purpose of noting the draft resistance of economizers. Owing to error of readings, the draft resistance of econo-

mizer being estimated as a difference of two draft observations, it is possible that the errors may be in opposite directions, tending to increase the actual draft resistance, or vice versa. Draft readings may be influenced by the velocity of flow of gases or by the relative heights of the two points of observation.

122 Gas analyses are usually taken at the boiler and not in the economizer setting, so that tests of record do not indicate closely the degree of excess air passing through economizer or the influence of air leakage in flues and economizer setting.

TABLE 5 CORRECTION FOR ALTITUDE; CHIMNEY CAPACITIES, OIL FUEL

Altitude, Feet above Sea Level	Normal Barometer	R Ratio Barometer Readings—Sea Level and Altitude	R^2 Ratio Increase Effective Stack Height	$R^{\frac{2}{5}}$ Ratio Increase Stack Diameter
0	30.00	1.000	1.000	1.000
1000	28.88	1.039	1.079	1.015
2000	27.80	1.079	1.164	1.030
3000	26.76	1.121	1.257	1.047
4000	25.76	1.165	1.356	1.063
5000	24.79	1.210	1.464	1.079
6000	23.87	1.257	1.580	1.096
7000	22.97	1.306	1.706	1.113
8000	22.11	1.357	1.841	1.130
9000	21.28	1.410	1.988	1.147
10000	20.49	1.464	2.144	1.165

123 Logically, the draft resistance of an economizer is proportional to the square of the velocity of flue gases, and to the length of economizer. The velocity of gases is directly proportional to the boiler horsepower, with an assumed excess air supply, and inversely proportional to the net area of the economizer, side dampers closed, and likewise proportional to the mean absolute temperature of gases within the economizer. The draft resistance of the economizer for sea level has thus been formulated as follows:

$$D_e = \left(\frac{\text{B.h.p.} \times T}{A}\right)^2 \times \frac{L}{T} \times \frac{J}{10^9} = \left(\frac{\text{B.h.p.}}{A}\right)^2 \frac{T \times L}{10^9} \times J \quad \ldots\ldots [18]$$

in which

D_e = draft resistance of economizer

B.h.p. = boiler horsepower actually developed, a quantity

TABLE 6—PART 1 APPROXIMATE DRAFT RESISTANCE, INCHES OF WATER: STANDARD GREEN FUEL ECONOMIZERS WITHOUT SIDE DAMPERS; 4 SQ. FT. OF ECONOMIZER HEATING SURFACE PER RATED B.H.P. OF BOILER.

BASED ON 50 PER CENT EXCESS OF AIR SUPPLY; FEEDWATER TEMPERATURE 150 DEG., SEA LEVEL, AND OTHER ASSUMED DATA. OIL FUEL.

Draft Resistances in Inches of Water Column for various Widths and Lengths.

Boilers at 100 per cent load of rated capacity.

Temperature gases entering economizer, deg. 425

Temperature gases leaving economizer, deg. 275

Average temperature in economizer, deg. 350

Gases per b.h.p. of boilers, lbs. 52.8

Assumed b.h.p. of economizer 3 b.h.p. per tube at rating ..... b.h.p. $= 3\ N - 3\ (L \times W) \times 1.00$

Area	16.6		21.85		27.00		32.25		39.25	
Wide	4		6		8		10		12	
L	B.h.p.	D''	B.h.p.	D''	B.h.p.	D''	B.h.p.	D''	B.h.p.	D''
8	96	0.000282	144	0.000365	....	.....	....	.....	....	.....
12	144	0.000951	216	0.00123	288	0.00144	....	.....	....	.....
16	192	0.00225	288	0.00292	384	0.00341	480	0.00373	....	.....
20	240	0.00440	360	0.00571	480	0.00666	600	0.00729	720	0.00708
24	288	0.00761	432	0.00989	576	0.0115	720	0.0126	864	0.0122
28	336	0.0121	504	0.0157	672	0.0183	840	0.0200	1010	0.0194
32	384	0.0180	576	0.0234	768	0.0273	960	0.0299	1150	0.0290
36	432	0.0257	648	0.0333	864	0.0388	1080	0.0425	1300	0.0413
40	480	0.0352	720	0.0457	960	0.0533	1200	0.0583	1440	0.0567
44	528	0.0469	792	0.0610	1060	0.0709	1320	0.0776	1580	0.0754
48	576	0.0609	864	0.0792	1150	0.0920	1440	0.101	1730	0.0990
52	624	0.0774	936	0.100	1250	0.117	1560	0.128	1870	0.125
56	....	.....	1010	0.125	1340	0.146	1680	0.160	2020	0.156
60	....	.....	1080	0.155	1440	0.178	1800	0.197	2160	0.191
64	....	.....	....	.....	....	.....	1920	0.239	2300	0.232
68	....	.....	....	.....	....	.....	2040	0.287	2450	0.279
72	....	.....	....	.....	....	.....	2160	0.340	2590	0.331
76	....	.....	....	.....	....	.....	2280	0.400	....	.....
80	....	.....	....	.....	....	.....	2400	0.467	....	.....

DRAFT RESISTANCE

CORRECTION FACTORS FOR INCREASED AREA OF SIDE DAMPERS

No. of Dampers Open	No. of Tubes Wide				
	4	6	8	10	12
0	1.00	1.00	1.00	1.00	1.00
One Side	0.484	0.564	0.621	0.667	0.769
Both Sides	0.285	0.361	0.423	0.476	0.581

APPROXIMATE CORRECTION FACTORS FOR 75 PER CENT AND 100 PER CENT EXCESS AIR SUPPLY

Per Cent Excess	50	75	100
At Rating	1.00	1.34	1.74
At 1.5 Rating	1.00	1.34	1.74

TABLE 6—PART 2 APPROXIMATE DRAFT RESISTANCE, INCHES OF WATER; STANDARD GREEN FUEL ECONOMIZERS WITHOUT SIDE DAMPERS; 4 SQ. FT. OF ECONOMIZER HEATING SURFACE PER RATED B.H.P. OF BOILER.

BASED ON 50 PER CENT EXCESS OF AIR SUPPLY; FEEDWATER TEMPERATURE 150 DEG., SEA LEVEL, AND OTHER ASSUMED DATA. OIL FUEL.

Draft Resistances in Inches of Water Column for various Widths and Lengths.

Boilers at 150 per cent of rated capacity or 50 per cent overload.

Temperature gases entering economizer, deg. 500

Temperature gases leaving economizer, deg. 330

Average temperature in economizer, deg. 415

Gases per b.h.p. of boilers, lb. 54.6

Assumed b.h.p. of economizer 3 b.h.p. tube at rating ..... b.h.p. $= 3\ N \times 1.50 \times 54.6 \div 52.8$

16.6		21.85		27.00		32.25		39.25		Area
4		6		8		10		12		Wide
B.h.p.	D"	B.h.p.	D"	B.h.p.	D"	B.h.p.	D"	B.h.p.	D"	L
149	0.000732	223	0.000951	....	.....	....	.....	....	.....	8
223	0.00247	335	0.00321	447	0.00374	....	.....	....	.....	12
298	0.00586	447	0.00759	596	0.00886	745	0.00972	....	.....	16
372	0.0114	558	0.0149	745	0.0173	931	0.0189	1120	0.0184	20
447	0.0198	670	0.0257	893	0.0299	1120	0.0328	1340	0.0318	24
521	0.0314	782	0.0408	1040	0.0476	1300	0.0520	1560	0.0505	28
596	0.0468	893	0.0609	1190	0.0708	1490	0.0776	1790	0.0754	32
670	0.0667	1010	0.0868	1340	0.101	1680	0.111	2010	0.107	36
745	0.0915	1120	0.119	1490	0.138	1860	0.152	2230	0.147	40
819	0.122	1230	0.158	1640	0.184	2050	0.202	2460	0.196	44
893	0.158	1340	0.205	1790	0.239	2230	0.262	2680	0.255	48
969	0.201	1450	0.261	1940	0.304	2420	0.333	2903	0.322	52
....	.....	1560	0.326	2090	0.379	2610	0.416	3130	0.406	56
....	.....	1680	0.401	2230	0.465	2790	0.512	3350	0.497	60
....	.....	....	.....	....	.....	2980	0.619	3570	0.601	64
....	.....	....	.....	....	.....	3160	0.745	3800	0.724	68
....	.....	....	.....	....	.....	3350	0.884	4020	0.859	72
....	.....	....	.....	....	.....	3540	1.037	....	.....	76
....	.....	....	.....	....	.....	3720	1.212	....	.....	80

Table 6 has been computed by the use of the following empirical formula:

$$D = \frac{P_o}{P_1}\left(\frac{\text{B.h.p.}}{A}\right)^2 \frac{T \times L}{10^9} \times J$$

in which

D = the loss in pressure through the economizer expressed in inches of water column

P_o = normal barometer, sea level

P_1 = barometer at the altitude of plant

B.h.p. = B.h.p. rating of standard economizer as assumed above

A = net area in sq. ft. between tubes for the flow of gases

T = average absolute temperature in deg. fahr. of the gases in the economizer

L = the length of economizer in rows of tubes

J = a coefficient determined from tests

For the purposes of this table the value of J is taken as 1.3, an average of values from a number of tests ranging from about 50 per cent above to 50 per cent below this figure. For 75 per cent and 100 per cent excess air supply, see approximate correction factors. If the volume of excess air due to leakage is sufficient to change the temperatures assumed, the correction factor for excess air will be correspondingly changed. B.h.p. refers to work done by boilers only, based on feedwater temperature entering the boilers, and not that entering economizers.

Draft resistance is based on areas of standard economizers with standard spacing and tube length. No allowance has been made for the resistance of any additional breeching or turns made necessary by the use of the economizer. Such resistances must be added to the values given in table.

For altitude D should be increased inversely with the barometric reading.

See footnotes, Table 7, regarding air supply greater than 50 per cent excess, leakage through brick settings, etc.

proportional to the weight of gases flowing per minute

A = net area through economizer, side dampers closed

T = mean absolute temperature, deg. fahr., of gases within economizer

L = number of sections of tubes in economizer

J = a constant, assumed as 1.3

124 The quantity D is, in reality, a draft coefficient based on the fundamental quantities and, neglecting J, should be modified by a correcting factor, depending on the designer's data. Based on the writer's data, this coefficient averages about 1.3 and varies from 0.65 to 2.0; J is therefore made 1.3.

125 Table 6 gives approximate resistances for various economizers, based on boiler horsepower ratings corresponding to 100 and 150 per cent load on boilers and economizers, and indicates the assumed temperatures entering and leaving the economizer, ratio excess air, etc. It is important to note that the boiler horsepower indicated represents the actual boiler horsepower developed by the boiler and not the rated boiler horsepower, a quantity for purposes of estimating the resistance of the economizer strictly proportional to the actual weight of gases flowing per second.

126 The fundamental equation [6] of draft resistance is modified, due to installation of economizers, by the introduction of an added term covering the resistance of economizers and a correction of chimney temperature. The equation [12] for boiler horsepower for Case III, central chimneys, is then written

$$\text{B.h.p.} = \frac{60^2}{G} \sqrt{d^5} \sqrt{\frac{fKH - (D_1 + D_2 + D_e)}{2.3\,T\,(H + 1.5d)}} \qquad [19]$$

where K is dependent on economizer outlet temperature.

127 Equation [19] is rendered approximate, as the term for breeching resistance is now made dependent on economizer outlet temperature; however, the breeching in a well designed economizer plant should be less than 10 diameters in length and these two errors may compensate. If added flue turns are introduced, then, for close figuring, [19] must be corrected accordingly.

128 Owing to the special nature of economizer installations, variation in size and arrangement of economizers, etc., it is not practicable to compute a general table of chimney capacities for economizer plants. If a chimney of minimum first cost be selected

for the given boiler conditions, ignoring the installation of economizers, then, to provide for economizers it should merely be necessary to increase the height and not the diameter of chimney to compensate for the added resistance, and lowering of stack temperature. This increase in height may be determined approximately, based upon the known increase in draft resistance and reduction of chimney temperature, from the draft tables. It is not possible to compute any simple table setting forth accurately the ratios of increased chimney heights for the various stated conditions, based on chimney capacities without economizers, by reason of the involved breeching resistances as well as stack resistances and varying ratios of total chimney draft required to overcome chimney friction.

129 It is a simple matter to compute for central chimneys a table of ratios giving the reduction of total boiler horsepower due to the installation of economizers, which, in connection with Table 4, enables a ready determination of chimney sizes. Table 7 has been computed on this basis and has reference to the capacities given in Table 4 for central chimneys without economizers.

130 It is self-evident that the minimum heights given in Table 4 will not furnish sufficient draft for any reasonable capacity, with the added friction and cooling of gases due to economizers. Table 7 indicates the minimum heights consistent with reasonable chimney proportions.

131 In order to obtain a chimney of minimum first cost when an economizer is used, trial and comparison should be made of several sizes.

GENERAL CONSIDERATIONS

132 The chimney capacities set forth in tables are conservative for the exact conditions stipulated, although the margin for safety is not so great as might ordinarily be presumed from the approximate nature of chimney calculations.

133 These tables will not have application to all types of boilers and all conditions of operation and refer specifically to boilers of the highest type and under favorable conditions of operation. If the designer is uncertain as to the conformity of his conditions to those stipulated in tables, due margins for safety should be allowed for selecting chimney sizes.

134 While the writer has obtained results of operation for

TABLE 7 SHOWING APPROXIMATE RATIOS OF CAPACITY OF CHIMNEYS WITH ECONOMIZERS TO CAPACITY OF SAME CHIMNEY WITHOUT ECONOMIZER BASED ON ASSUMPTIONS BELOW

	At Rated Load									150 Per cent of Rated Load								
Initial temperature of gases in stack without economizer, deg.	425									500								
Initial temperature of gases in stack with economizer, deg.	275									330								
Db. Draft at chimney side of boiler damper, in.	0.15									0.30								
De	0″	0.005″	0.010″	0.025″	0.05″	0.10″	0.15″	0.20″	0.30″	0″	0.005″	0.010″	0.025″	0.05″	0.10″	0.15″	0.20″	0.30″
H	Ratio of Chimney Capacities									Ratio of Chimney Capacities								
60	0.502	0.439	0.365															
70	0.669	0.638	0.608	0.500	0.233													
80	0.737	0.717	0.696	0.630	0.500													
90	0.774	0.759	0.743	0.694	0.605	0.364				0.495	0.455	0.411	0.233					
100	0.798	0.786	0.774	0.735	0.667	0.498	0.233			0.645	0.623	0.600	0.526	0.371				
110	0.815	0.805	0.794	0.762	0.705	0.575	0.406			0.715	0.699	0.684	0.634	0.542	0.275			
120	0.827	0.819	0.810	0.782	0.734	0.628	0.497	0.321		0.759	0.748	0.736	0.698	0.630	0.465	0.189		
130	0.837	0.830	0.822	0.798	0.756	0.665	0.560	0.428		0.788	0.778	0.769	0.738	0.684	0.561	0.401	0.080	
140	0.844	0.838	0.830	0.809	0.772	0.693	0.603	0.498	0.126	0.809	0.801	0.795	0.767	0.721	0.623	0.504	0.347	
150	0.851	0.844	0.838	0.818	0.785	0.715	0.637	0.548	0.299	0.825	0.818	0.811	0.789	0.750	0.666	0.571	0.456	
160	0.855	0.850	0.844	0.826	0.797	0.733	0.664	0.586	0.388	0.837	0.831	0.825	0.805	0.771	0.699	0.619	0.526	0.254

The capacity ratios given above are applicable to chimney capacities given in Table 4 for boiler plants without economizers, chimneys centrally located sea level, and are subject to all of the limitations stated for Table 4. *De* = draft resistance due to economizer, for values of which see Table 6. If economizer layout introduces greater breeching resistance than assumed in development of Table 4, *De* should be increased accordingly to cover additional length of and turns in breeching. Table 4 is based on straight run of breeching. B.h.p. refers to work done by boilers only, based on feedwater, temperature entering boilers, and not that entering economizers. Boiler efficiencies are as stated in Table 4, not including economizer saving. Ratios given for various heights enable the maintenance of same net draft at boiler outlet for stated boiler ratings, making necessary allowance for loss of draft due to cooling of chimney gases and interposed economizer resistance. For sake of uniformity with other tables, ratios are based on 50 per cent excess air supply at maximum load measured in chimney. This condition is realizable in small plants having masonry settings and in large plants with steel breechings and steel economizer casings. For large plants, with brick flues and brick economizer settings, air leakage may increase relative volume of chimney gases from 20–40 per cent or more. *De* should be modified accordingly for resistance, and the then calculated B.h.p. of chimney reduced to allow or air leakage. Correction must also be made for modified chimney temperature.

numerous chimneys burning oil fuel, it is generally the rule that oil-burning chimneys in actual service are underloaded and that comparatively few chimneys are operating at their maximum capacity. Therefore, in computing chimney capacities given it has been the writer's aim to determine from test observations the actual boiler resistances for the various conditions of operation, and from this starting point to calculate chimney height sufficient to produce the required draft at boiler, and diameter sufficient to carry away the necessary volume of gases, with the margin of draft available for the friction of flow through breeching and chimney.

135 Where building conditions demand chimneys of considerable height the draft required at the boiler outlet becomes a smaller proportion of the potential draft of the chimney, and accordingly an error in the assumption of the draft resistance of the boiler will have less effect on the total capacity of the chimney of given dimensions.

136 When selecting chimneys of reduced height for boilers operating at or near rating the draft resistance of the boiler for least cost and economical operation requires a larger proportion of the potential draft of the chimney. An error in the assumption of draft resistance of the boiler largely affects the capacity possible with a chimney of given dimensions.

137 In all instances the diameter of the chimney and frictional loss within the chimney and breeching have been based on 50 per cent excess air over chemical requirements. For a number of years the writer thought it necessary to provide for not less than 100 per cent excess air; but from actual experience and data from a number of tests where the firing conditions were directed by ordinary operators, it has been shown that 50 per cent excess air is sufficient for good practice under maximum load conditions in first class installations.

138 The various features affecting the draft resistance of a boiler have been pointed out, many of these being subject to variation by the local operator. The greatest influence on draft resistance, however, is that due to excess air. Under ideal conditions it is possible to fire boilers with approximately 10 per cent excess air and even less, giving a corresponding reduction in the draft resistance of the boiler. As the excess of air supply is reduced, the capacity of a chimney increases in a multiplied ratio, due, not only to the decreased volume of gases to be handled and

the decreased weight of gases per b.h.p. by reason of greater boiler efficiency, but also to decreased draft resistance of the boiler and the greater draft available for the flow of gases through breeching and chimney. It will therefore be possible not only to point to chimneys of the dimensions given in tables showing a greater capacity than stated corresponding to reduced air supply, but also of less capacity than stated corresponding to increased air supply.

139 In considering boilers of differing types, the chimney dimensions will depend largely on the boiler design and efficiency. particularly with respect to the draft resistance of the boiler and the minimum air supply as dependent on furnace design. Certain types of boilers will show less draft resistance and others more than indicated for Babcock and Wilcox boilers, with a corresponding variation in chimney capacity.

140 Many inexperienced in the burning of crude oil carry the impression that the mere closing of the damper will result in the reduction of excess air supply practically to the theoretical limit. It is actually possible to cause the formation of smoke with considerably more than 50 per cent excess air supply in a poorly designed boiler furnace. The furnace must have a particular shape, volume, and arrangement of checkerwork with respect to the burner to permit the complete admixture of air for combustion and vaporized oil; and the furnace efficiency with respect to this admixture, influenced by the surrounding heated brickwork, determines the minimum excess air permissible without undue smoking.

141 In the calculation of chimney tables it was deemed best not to provide sufficient margin for safety in order to cover all possible conditions of operation, but rather to establish as a benchmark the maximum chimney capacities based upon the draft resistances obtained during conditions of uniform load, and to modify capacities as need be for the special conditions of operation, which should be carefully studied before selecting chimneys for a given installation.

142 When operating with 50 per cent excess air, uniform load, and furnaces properly adjusted, there should be a slight haze from each boiler; on uniform load there should not be an objectionable amount of smoke from central chimneys connected to a number of boilers, except when fires are unequally adjusted.

As is well known, automatic control of fires insures an equal division of load among all boilers in service.

143 It is possible for short intervals to force the chimneys beyond their maximum capacity, giving rise to increased smoke density; but good practice does not sanction such heavy smoking, except for the inevitable short intervals on swinging loads.

144 In passing, it may be stated that it is common practice to gage the air supply roughly by the density of smoke emitted from the chimney, and it is wise in boiler plant design to provide openings through the roof so that the firemen can conveniently observe the chimneys.

145 Tables provide for the slight reserve necessary to regulate fluctuating steam pressure with careful firing, but do not provide sufficient reserve to restore steam pressure rapidly after a material pressure drop. During such intervals careless firing may require a very large momentary overload on boilers. While a certain draft margin is necessary, a chimney which makes necessary careful attention of furnaces is a positive benefit from the standpoint of boiler abuse and cost of tube renewals.

146 If a number of boilers be connected to a central chimney subject to hand-firing, certain boilers under unequal adjustment of fires will be operating at overload and others at underload. To offset these conditions and to maintain the maximum total capacity required, it is necessary slightly to increase the height of stack, and in the table no margin is specially provided to cover this contingency. When firing under automatic control, which insures an equal division of load among all boilers in service, such margin for increase is unnecessary.

147 It has been found in large plants having central chimneys and subject to heavy swinging loads, that on a sudden increase of load there is a sluggishness of chimney action due to inertia of gases within the chimney and breeching, as well as a time element required for the heating of the chimney; smoke is usually formed during such intervals. Particularly where flues are of masonry construction it is desirable to increase the height of chimney slightly to counteract this influence, although occasional smoking cannot even then be completely eliminated.

148 For swinging loads the writer would reduce the tabulated capacities for central chimneys from 10 per cent to 20 per cent, this factor being one of judgment and not based on experimental data.

149 When boilers have been operating for a period at reduced load, on a sudden increase in load owing to the cooling action of the furnace, an increased air supply will be required over uniform load conditions in order to obtain the desired capacity without smoking. This condition will continue until the furnace brickwork has been thoroughly heated, and common practice permits smoking during this short interval.

150 Consideration should be given to a certain reserved draft in order to insure, at all loads, a positive draft within the setting by throttling the air admission. It has been pointed out by Dr. Jacobus that under certain load conditions, should a negative draft within the setting occur, there will be a tendency for the hot gases to pass outward through the setting making possible a considerable overheating of the brick furnace. On the contrary, with a positive draft within the setting, a certain amount of air infiltration will tend to cool the brickwork and thus increase the life of the setting. In general, the tendency to form a negative draft diminishes with overload.

151 In the design of chimneys it is necessary to give careful consideration to the question of minimum air supply by reason of the tendency of increased smoking with changes of load when operating near the theoretical limit. Sooting of boiler tubes might easily reduce the heat transfer of the boiler to such an extent as to more than offset the economy due to decreased air for combustion.

152 A well designed central chimney subject to variation of number of boilers in service requires, for maximum economy, a comparatively small amount of stack and breeching friction in order to insure what might be termed "draft regulation." In boilers subject to hand adjustment or, for that matter, automatic control of air supply, it is desirable to have a nearly fixed draft at the boiler outlet for each addition of load, with the least possible change due to variation in the number of boilers in service. It is self-evident that this variation in draft will give a variable air supply, with a fixed position of damper for each load, and thus reduce the fuel economy of the plant.

153 The proper determination of chimney height, with respect to the maximum draft required at boilers, will have a marked influence on the fuel economy of the plant. While a certain draft margin is necessary for the special conditions of plant operation, it should at all times be the aim to provide a chimney

with the least possible excess draft over the necessary maximum.

154 In plants subject to uniform factory or pumping loads approximately at rating, the chimney height should be very much reduced and such that when boilers are operated with nearly wide open dampers the air supply will not materially exceed the requirements for economical firing.

155 With excessive height of chimney it is not possible to open the damper more than a fractional amount, thus giving a large increased air supply for a very small added opening of damper, making economical operation at variable loads impossible with hand control, and more difficult with automatic control.

156 The influence of excessive draft becomes more apparent when boilers are operated for intervals at reduced capacity. If the firemen be careless and leave the dampers and ash doors unchanged during variations of load, it is apparent that the draft resistance of the boiler becomes nearly a constant, and except for a slight decrease in the temperature at reduced loads, the tendency is toward a practically constant flow of gases through the setting regardless of the rate of firing, and correspondingly, an increasing ratio of excess air at light loads. While with tall chimneys the excess air supply is abnormal even at maximum loads, it becomes thus increasingly wasteful on reduced loads.

157 In one instance which came to the writer's observation, a chimney gave 300 per cent excess air when operating at rating, and 450 per cent excess air when operating 30 per cent load, the dampers and ash doors being wide open in both instances. In the latter instance the resulting boiler efficiency was less than 40 per cent, whereas, even at this reduced load, the efficiency should have been approximately 75 per cent.

158 Experience has indicated that plants having tall chimneys subject to hand control are, in general, uneconomically operated; while plants having relatively low chimneys indicate a superior performance, due largely to the decreased margins for excess draft and possible wastage of fuel with excess air supply. This fact is particularly impressive after an examination of the economy attainable in power plants in connection with tall office buildings, where high chimneys are a necessity. In such instances a reduction of stack diameter to absorb by stack and breeching friction a considerable portion of the excess draft will result in an improved economy of plant.

CHIMNEYS FOR AUXILIARY STEAM POWER STATIONS

159 It has been pointed out that chimneys of reduced height are possible under certain conditions of operation, namely, with uniform load, approximately at rating, but the impression should not prevail that tall chimneys are unnecessary for oil firing under certain conditions of heavy load.

160 In auxiliary steam power stations it will be found necessary at times to force boilers to approximately 100 per cent overload. Under these conditions, due to the rapid increase of draft resistance of boiler, the heights of central chimneys will vary, depending on the type and size of boiler, plant arrangement, etc., from less than 100 ft. to about 150 ft. for sea-level conditions, and even greater heights will be required at altitudes.

161 The height may be somewhat reduced below that otherwise necessary on heavy overloads by increasing judiciously the admission area through floor of furnace. On large boilers reduction of draft and chimney height at overloads may also be effected by the provision of larger air admission doors, limited, however, by the creation of a negative draft within setting, as already discussed.

162 It is important to note that the chimney heights stated for certain overload capacities apply only when a sufficient number of the connected boilers are in operation to maintain the stated initial chimney temperature, and to prevent undue cooling within the chimney. In auxiliary steam power plants, or other stations where at times comparatively few of the boilers may be in operation on a central chimney, this feature must be considered.

163 If the temperature of the water is, say, 100 deg. fahr., or less, within a boiler connected to a cold chimney, then, on an emergency call, a considerable period will elapse before the gases entering the chimney will reach such temperature as to produce an appreciable draft and make possible rapid firing. If, as is ordinarily the case, all the boilers be kept under steam by intermittent and alternate firing, the temperature of the boiler heating surface will be considerably above the boiling point, and the mere opening of the boiler dampers will cause a flow of air through the setting, the incoming air being heated by the hot water within the boiler tubes. The starting of the fire will grad-

ually increase the temperature of the escaping gases, but owing to the reduced draft, a considerable interval of time will elapse before the boiler will be operating at such rate of firing as to give the necessary increase in temperature of escaping gases and chimney draft for any appreciable overload on the boiler. Under these conditions it is evident that the taller the chimney the less the required time for getting under way; also, that an independent chimney will, in general, enable quicker starting than a central chimney when firing a fractional number of boilers. On the other hand, if a central chimney be employed, and one or more boilers are constantly under fire, the chimney will at all times have a relatively higher minimum temperature, and by reason of the draft produced, enable the quicker starting up of a large number of connected boilers than is possible with independent chimneys.

164 As service demands become more exacting it is probable that auxiliary steam power stations will ultimately be provided with forced draft fans to supply air for the rapid starting of boilers which are to be used for the first few minutes, and until boilers, furnaces, flues and chimneys are thoroughly heated, thus following, to a certain extent, battleship practice; or steam jet chimney blowers may be used while starting.

EXAMPLES FROM PRACTICE

165 *Plant No. 1.* A plant of three Babcock and Wilcox boilers, 7 high, total 275 h.p.; one Green fuel economizer 6 by 20; one side damper; one pair of induced draft fans; one 42-in. discharge stack; 54-ft. height above floor line. Plant initially installed for coal burning. When fired with oil fuel, at rating and at slight overloads it was found that the natural draft from chimney was sufficient to produce the necessary draft at boilers to overcome resistance of economizers and resistance of blades of fan wheels with fans at rest. When a wind storm reduced the height to 40 ft. the capacity was still ample. A second wind storm reduced the height to 19 ft. which was ample at light loads, but required assistance of fans at 180 r.p.m. at rating.

166 *Plant No. 2.* A plant having a number of 9 high Babcock and Wilcox boilers; after the San Francisco earthquake of 1906, was able to operate at rating with chimney height reduced by earthquake to 25 ft. above floor line.

167 *Plant No. 3.* A plant of 12 high Babcock and Wilcox boilers, with horizontal short breeching and steel chimney 115 ft. above floor line; sea level; chimney diameter relatively large and friction small; three boilers connected to one chimney, one boiler, on test, operated at 210 per cent of rated capacity, test conditions.

168 *Plant No. 4.* A plant of 12 high Babcock and Wilcox boilers; a total of 15 boilers, connected to a central chimney 125 ft. above floor line; Green fuel economizers; induced draft fans; one boiler operated at 208 per cent of rating, other boilers at about rating. The ordinary draft of chimney was insufficient, and necessitated the use of fans. Economizers 80 tubes long, resistance excessive.

169 *Plant No. 5.* A plant of 12 high Babcock and Wilcox boilers, 8 boilers connected to a central chimney; Green fuel economizers; panel system. Chimney reduced by earthquake to 70 ft. Natural draft sufficient for operation of boilers at about rating, but insufficient for overload.

170 *Plant No. 6.* A plant of 14 high Babcock and Wilcox boilers; central chimney; large diameter; approximately 70 ft. height above floor line. Maximum capacity one boiler under test, when other boilers operating at about rating, natural draft 145 per cent of rating.

171 *Plant No. 7.* Stirling boilers, 250 h.p., connected to independent overhead chimney 80 ft. height above floor line. Maximum recorded capacity, but not maximum possible capacity, 90 per cent overload, still showing an excess air supply; plant operating factory load, boilers intended for operation at rating only; on account strong draft from chimney could not get fireman to operate boilers at rating. Finally reduced chimney height to 40 ft. to prevent such excess air through furnaces as would prevent excessive overloads on boilers. In this manner capacity considerably reduced, but still able to obtain greater capacity than rating. Exact overload not determined. Plant economy increased by lowering the temperature of escaping gases and amount of excess air.

172 *Plant No. 8.* Battery of two Stirling boilers 250 h.p. each, connected to a central overhead chimney with A-shaped breeching, the total height being 40 ft. above floor line. Intended for use in oil fields with unskilled firemen. Height fixed at 40 ft. to prevent overload on boilers and the waste of fuel from excessive overload and from excess air. Boilers have operated at capacities in excess of rating.

173 *Plant No. 9.* Office building plant. Stirling boilers, chimney height approximately 200 ft. on account of excess draft even with partially open dampers and air doors excess air over 200 per cent at rating, probably 15 to 20 per cent waste of fuel due to excess air.

THE INTERNATIONAL SMOKE ABATEMENT EXHIBITION

HELD IN LONDON, MARCH AND APRIL 1912

By George H. Perkins

ABSTRACT OF PAPER

The solution of the smoke problem must necessarily rest with the mechanical engineer. The development of a strong and intelligent public sentiment on the economic and sanitary necessity for smoke abatement, however, must come through the widespread campaign of education in which the Society should take an active part. Societies for this specific purpose are doing much effective work in England with most excellent results. The lines along which these organizations are working are briefly indicated in this paper, which is an informal report made originally to the trustees of the Lowell Textile School, Lowell, Mass., who delegated the writer to attend the International Smoke Abatement Exhibition and Conference held in London from March 23 to April 4, 1912.

THE INTERNATIONAL SMOKE ABATEMENT EXHIBITION

HELD IN LONDON, MARCH AND APRIL 1912

By George H. Perkins, Lowell, Mass.

Associate Member of the Society

An International Smoke Abatement Exhibition and Conference, under the auspices of the London Coal Smoke Abatement Society, was held at the Royal Agricultural Hall, Islington, London, March 23 to April 4, 1912.

2 The object of the exhibition was to make a comprehensive display of the most approved modern methods and devices for the abatement of coal smoke, applicable either to industrial plants or domestic fires. There were also exhibits showing the injurious effects of smoke deposits upon health, building materials, works of art and vegetable life, with the view of creating an active and intelligent public sentiment on the importance of this widespread nuisance.

3 The educational work was supplemented by a series of conferences which were well attended by official delegates from most of the principal cities of England and Scotland as well as from Germany, Holland, Sweden, and other foreign countries.

4 The exhibition attracted much favorable attention from the press and general public, and upon the whole was considered a success. The great coal strike which was at its height at the time, while interfering with the attendance and delivery of exhibits to some extent, tended to increase public interest in the affair and to divert attention to the larger question of the conservation of the coal supply.

PRESENT STATUS OF THE SMOKE PROBLEM IN ENGLAND

5 Smoke has been a recognized nuisance in England for nearly a century. As early as the year 1819, Parliament ap-

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

pointed a committee to investigate this question which reported that effective smoke abatement devices existed and should be used. The public health acts of 1875 and 1891 included sections covering smoke abatement which are the only present laws on the subject. These statutes have not proved satisfactory as they are not sufficiently definite and do not make provision for their proper enforcement.

6 Practically all of the effective work done in abating the smoke nuisance has been accomplished during the past 15 years. The great improvement in atmospheric conditions, as far as London is concerned, has been due principally to the following agencies:

a The electrification of all of the underground railways formerly operated by coal burning locomotives.

b The rapid increase in the use of gas appliances for heating and cooking. When authorities agree that more than one half of London's smoke comes from domestic fires burning soft coal, the following figures are significant:

Year	Number of gas appliances in use
1891	46,000
1896	223,000
1901	445,000
1906	989,000
1911	1,494,000

c The work of the London Coal Smoke Abatement Society, a voluntary association which has for its objects the following:

1 To aid in enforcing the existing law through the local sanitary authorities, by the employment of inspectors who, after investigation, shall report offenders to the proper officials.

2 To promote and encourage all voluntary efforts to abolish smoke from private dwellings and to investigate the best means for effecting this object.

3 To obtain particulars and evidence of methods of dealing with smoke nuisances at home and abroad.

4 To publish information on matters relating to smoke nuisances, stimulate invention through the offer

of prizes and to conduct tests on heating, cooking and stoking apparatus.

5 To effect the amendment of the present laws with the object of making them more efficient.

7 Other societies of the same character have been organized in some of the larger cities and are doing effective work along similar lines to that of the London society. One of the most active of these is the Smoke Abatement League of Great Britain which has its headquarters in Manchester and branches in many of the other large industrial centers.

8 The most direct evidence of the improvement made in smoke abatement in recent years is in the record of observations of atmospheric conditions taken in many of the large cities. The "black fogs" once so prevalent in London and which have been proven to have been due largely to smoke are to-day practically unknown. The winter sunshine of London is to-day about 40 per cent of that observed in the country districts, which is a figure double that of 30 years ago.

9 In nearly all of the larger cities a marked improvement along similar lines has been made each year and through the agencies already mentioned. In the manufacturing districts particularly, the boiler users are taking increasing interest in this matter, realizing that it has an important influence upon the efficiency of their plants. The public is also aroused to the situation and the urgency and practicability of smoke abatement seem to be generally appreciated.

SUMMARY OF PAPERS PRESENTED AT THE CONFERENCE

10 The papers presented at the conference may be grouped under the following heads:

A Smoke Pollution
- 1 Economic and Artistic Aspects
- 2 Effects on Plant Life

B Smoke Abatement
- 1 Organization of Preventive Action
- 2 Physical Principles of Smoke Abatement

C Laws and Legislation
- 1 Proposed New Legislation
- 2 Administration of Existing Law

(*A*-1) *Economic and Artistic Aspects*

11 Under this head papers were read treating of the effects of smoke upon building materials, mural and house painting, ornamental and structural metal work and also the measurement of soot deposits.

12 Examples were cited of the disintegration of the stone in various buildings due to sulphuric acid in the atmosphere. This acid is formed from the union of the oxidized sulphur compounds, resulting from the combustion of coal, with the moisture of the atmosphere. An analysis of stone taken from St. Paul's Cathedral showed 74 per cent of calcium sulphate, the original stone being calcium carbonate or limestone. Similar results were obtained on samples taken from Westminster and other well known and historic structures. Limestone appears to suffer more than any other variety of stone while granite is not seriously affected.

13 The effect of smoke on mural and fresco paintings was due to a similar action, the plaster beneath the pigments losing much of its binding power by the chemical change taking place. The effect upon house paint is also very pronounced, particularly on the lead paints. Practically all outside work must be repainted at least once a year. Zinc oxide paints were shown to be more stable than lead pigments against the action of atmospheric acids.

14 The most serious effect upon metal work is the corrosion of structural steel. Several collapses of iron roof trusses, notably that of the Charing Cross Station, have resulted from this cause. In some instances it was found that about 10 per cent of the iron had been dissolved into ferrous sulphate. The corrosion of uncovered iron or steel wire has also been observed and found to be extensive.

15 During the past year the soot fall of London has been carefully measured by means of specially devised soot gages. This work has been in charge of the officers of the Smoke Abatement Society and the extraordinary results obtained may be summarized as follows:

16 The total yearly deposit from the atmosphere was 650 tons per square mile, or a total of 76,050 tons per annum for the entire administrative county of London of 117 square miles. This figure includes 8000 tons of sulphates, 6000 tons of ammonia and 3000 tons of chlorides, the balance being carbon and tarry prod-

ucts. The deposit per square mile at Surrey, on the border of the metropolitan area, was only 195 tons per year, or less than one third that of London proper, showing clearly the comparative purity of country air.

(*A*-2) *Effects on Plant Life*

17 Evidence was given by the authorities in charge of the Kew Gardens and other parks as to the difficulty of maintaining vegetation in smoke affected districts. Only a few hardy plants and shrubs will survive the winter season in the large cities. The main effects of smoke on vegetation are due to the following:

1 Reduction of light and heat from the sun.
2 Soot deposits excluding still more light.
3 Tarry deposits blocking the pores of the plants.
4 Acid deposits lowering the vitality of plants.

18 The effect upon health of a smoke polluted atmosphere was clearly shown by figures given from Glasgow, where a study has been made of the effect of smoke fogs upon the death rate from bronchial diseases such as bronchitis, pneumonia and pleurisy. The deaths from these diseases alone increased at a remarkably rapid rate during prevalent fogs and could be traced to no other source.

(*B*-1) *Organization of Smoke Abatement Work*

19 The work of the various smoke abatement societies was described in detail and many suggestions were made regarding the possible future study of this problem. The majority of these suggestions were along the line of determining the actual economic loss to the community caused by smoke, it being generally conceded that the engineering side of the question is well in hand and sufficiently well developed to meet practically all conditions provided careful study is made of the factors involved.

20 Some of the elements in the economic loss to a city are as follows:

1 Added cost of artificial illumination.
2 Added cost of painting exteriors and interiors.
3 Added cost of laundering and other cleansing.
4 Damage to goods in stores and factories.
5 Effect of gloom upon the efficiency of workers.

21 Recognition was made of the splendid work done on smoke abatement in Cleveland, Ohio, and Chicago, Ill., where the progress made within the past five years has been remarkable. In

both of these cities the problem is handled by efficient departments of the municipal governments furnishing a most excellent example of honest, effective and thorough public service. These two cities are without question well in advance of any other American municipalities in the matter of smoke abatement.

(*B*-2) *The Physical Principles of Smoke Abatement*

22 The papers presented under this head covered various phases of the subject, including the effect of smoke upon steel making, losses due to incomplete combustion, hand and mechanical firing, the manufacture of smokeless fuels and the construction and operation of producer gas plants. The production of steel without objectionable smoke was held to be practicable, without affecting the quality of the steel.

23 Results of tests were given showing the loss due to unburned combustible matter in smoke and that due to a non-conducting layer of soot upon the surfaces of boiler tubes and plates.

24 The value of trained firemen was especially emphasized and the subject of hand firing and conditions under which it might be successful were discussed. It was proposed to add to the present license for firemen, a certificate of competency based upon a knowledge of proper conditions for perfect combustion, heating value of fuels, etc.

(*C*) *Laws and Legislation*

25 The larger part of the matter presented under this head would not be applicable in this country for obvious reasons. A strong plea was made, however, for uniform and definite legislation on smoke abatement. The principal paper in this group urged the appointment of a royal commission to inquire generally into the subject of smoke emission for the purpose, if possible, of having a stringent general law passed by Parliament. It was suggested that this royal commission, if appointed, be directed to inquire:

1 What standard, if any, is desirable as to the color or density of the smoke which should be deemed a nuisance?

2 How the color or density of smoke can best be identified?

3 Are the fines at present inflicted upon offenders sufficiently adequate to act as deterrents?

4 Who should be punished, the owner of the works, the

engineer in charge, or the stoker, when smoke nuisances result from the careless manipulation of well equipped installations?

5 Should any industries receive special treatment?

6 Should Government inspectors be appointed to act in regard to the emission?

7 Should Government premises be placed under the obligation to prevent smoke nuisances?

8 Ought any hourly or other time limit be fixed during which black or other smoke may be permitted to issue from factory chimneys?

9 Should all cases be heard by stipendiaries? (Paid magistrates.)

10 Can the issue of an unreasonable quantity of smoke be prevented except in cases of accident or breakdown?

11 Can it be shown that the installation of proper appliances generally results in economy as is contended by "smoke abaters"?

12 The incidence, intensity and duration of fogs and their effects on health.

13 The influence of smoke on health.

14 The damage of smoke to buildings, works of art and property generally.

15 Foreign efforts to cope with the smoke nuisance.

16 How far is it possible by smoke abatement to conserve the coal supply?

17 How far is it practicable to deal with the smoke from private dwelling houses?

CLASSIFICATION OF EXHIBITS

26 The exhibits may be classified into the following groups:

a Boiler furnaces; special grate bars and furnace construction; mechanical stokers of all types; smoke preventing devices of various kinds; fuel economizers; briquette making machines.

b Suction gas plants; gas and oil engines.

c Open coal-fire grates; stoves and ranges of all kinds; draft regulators and other smoke preventing appliances for domestic fires.

d Gas stoves and grates; gas heating and cooking appliances.

e Electric heating and cooking appliances.

f Testing devices, including anemometers, tintometers, soot gages, automatic gas analyzing devices, etc.

g Fans and other appliances for the removal of dust of various kinds from industrial plants.

h Smokeless fuels, both natural and artificial.

27 While there was little essentially new in the principles underlying the various devices exhibited, there has been a marked improvement made within a few years in their design and mechanical construction. There was also evidence of an appreciation and careful study of the theoretical side of the question which has undoubtedly contributed largely to the relatively better results obtained in recent years. The exhibits were so numerous and varied that only the most important will be mentioned.

MECHANICAL STOKERS

28 The stoker which appeared to be considered the best adapted for the prevailing conditions in England is the chain grate type.

29 When used with the proper fuel, preferably bituminous slack, this type appears to meet the requirements of smokeless combustion more nearly than any other device, particularly with water tube boilers. The best sample of this stoker exhibited was by the Babcock and Wilcox Company.

30 A number of types of coking stoker were shown, the best being those made by J. Hodgkinson of Salford and E. Bennis of Bolton. These stokers are better adapted to the Lancashire type of boiler, since they are not readily forced. In their action a reciprocating ram, mechanically driven, and located in the bottom of the hopper, forces a small quantity of fuel on to a "dead" or coking plate at each forward stroke. While the fuel remains upon this plate the volatile products are driven off and the coked coal then drops upon the grate bars which are of a sliding type and actuated by a cam motion on the front end. The mass of the fire is constantly carried toward the rear of the furnace by an intermittent and simultaneous movement of all the bars, the return stroke of each bar being independent.

31 There were a number of forms of sprinkler stoker shown

which, although of ingenious design, have not proved satisfactory where smokeless operation is desired.

32 While there were no underfeed stokers exhibited, there are a number of English machines of this class giving satisfaction under the proper conditions. The best known are the Vickers, Bennis, Underfeed and the Proctor. This type of stoker is not so well adapted for use with the average fuel available in England and is not so extensively used as in America.

PATENT FURNACES

33 There were many exhibits under this head, only a few of which possessed real merit. The majority of these appliances consisted simply of properly controlled steam jets with special forms of grate bars. There were a few good devices for the automatic regulation and timing of the supplementary air supply, both over the fire and below the grate. The best of these was the Kowitzke furnace which was excellently designed and simple in operation.

34 Various types of furnace construction, designed to effect a better mixing of the gases, were also shown. The Gregory patent furnace makes use of a supplementary jet of vaporized fuel oil ejected over the fire, which is claimed to add materially to the efficiency.

35 A special boiler known as the Bettington is designed to allow the use of a mixture of air and pulverized fuel or coal dust, introduced by air blast at the bottom of the boiler which is of the vertical type. This boiler is intended for use at mines or wherever fuel of a finely divided character is available.

MISCELLANEOUS DEVICES

36 A device for the optical analysis of the gases of combustion by means of an interferometer was shown by Carl Zeiss and Company. It is based upon the principle that the refractive power of the gases is dependent upon the content of carbon dioxide. A gas refractometer is so calibrated that the analysis of the gas may be read on the scale by simple inspection. A special form of tintometer was also shown in a portable form for gaging the density of smoke or fog.

37 The numerous forms of appliances adapted for domestic

uses, all showed improved features, but need not be described here.

38 Other interesting exhibits included "Coalite" and other artificial smokeless fuels, spark and soot catching chimney cowls and special chimney construction for effecting a dilution of the smoke as it is emitted.

CONCLUSIONS ON SMOKE ABATEMENT

39 In judging of the merits of the various types of mechanical firing appliances exhibited, it should be recognized that all of these devices were designed for local conditions and for burning English or Scotch coals. These fuels cost from $2.50 to $3.50 per ton at the plant, contain from 30 per cent to 35 per cent volatile matter and 11,000 to 13,000 b.t.u. per lb. In these respects they correspond to our Illinois or other western coals. This class of fuel is not available in New England for economic reasons. The freight charges on coal in this section amount to about $3.00 or nearly three quarters of the average cost of fuel per ton. It is therefore not feasible or economical to burn any but the highest grade of bituminous coal in this part of the country for industrial purposes. This class of fuel should contain about 14,000 b.t.u. per lb. and not over 20 per cent volatile matter. On the chain grate, for example, which gives excellent results in England, this coal could not be burned satisfactorily. It would be impossible to maintain a proper ignition temperature with such a low per cent of volatile matter without an excessive draft which would give large losses to the stack. The chain grate is meeting with success in this country wherever fuel of the lower grades is available.

40 Of the other devices, perhaps the most important are those automatically controlling the air supply, both over and under the fire. These eliminate the personal factor of the fireman and should be effective on any type of hand-fired furnace.

41 In conclusion, smoke abatement may be best effected in the present state of the art of fuel burning by thorough consideration of the following conditions. Careful attention to one or all of these factors by competent engineers will increase the efficiency of any plant and with efficient combustion comes smokeless operation.

1 Selection of a suitable fuel with provision for maintaining same at fixed standard of heat value.

2 Careful scientific study of the conditions prevailing in the plant, including draft, composition of gases, temperatures, etc.

3 Design or selection of type of furnace or apparatus best adapted to meet these conditions.

4 Proper construction or installation of same.

5 Careful selection of operating force.

6 Suitable instrumental aids or guides for the firemen and responsible engineer.

7 Frequent and thorough inspection to insure maintenance of highest possible efficiency.

WORK OF THE HAMBURG SMOKE ABATEMENT SOCIETY

42 Among the delegates present at the London conference was Herr Nies, chief engineer of the Vereins für Feuerungsbetrieb und Rauchbekämpfung, of Hamburg, who later extended to the writer in Hamburg many courtesies which brought him into contact with the work of the society.

43 The Hamburg society, which was organized in 1902, is essentially different from other organizations of this character in the following important features:

a It is an entirely voluntary organization of manufacturers and other boiler users bound together only by a common desire to obtain greater efficiency and less smoke from their power plants.

b The working staff of the society is composed wholly of technically trained engineers.

c All boilers owned by members are under the systematic and scientific control of the expert staff of the society.

d All members benefit mutually by the published results of all tests and investigations made by the engineers.

e The economy of smokeless combustion is the cardinal principle and the members have been convinced of this in a most practical manner, in the savings effected in fuel consumption by their co-operative effort.

44 The society now numbers 436 members, representing 1381 boilers. The annual dues of members are small, being only $5 and $5 additional for each boiler in their plants. The society is entirely self-supporting and draws its funds from three sources, annual dues, payment for special work or reports, and payments for outside work.

45 The education and control of firemen in the proper performance of their duties are also undertaken by the fireman instructors on the staff of the society.

46 Comparative tests of fuels, smoke preventive devices, etc., are carried out by the engineers and the results are circulated among the members. Advice is also given on purchase of fuel, the buyers being educated by the engineering staff to appreciate the importance of this matter.

47 The engineering staff of the society consists of a chief engineer, four assistant engineers and eight trained firemen who act as fireman-instructors. For the chemical and thermal analysis of fuels, the society avails itself of the testing laboratory of Dr. Aufhauser at Hamburg. This laboratory, which is one of the most complete of its kind in Germany, tests about 2000 samples of fuel annually. Dr. Aufhauser also edits a periodical called "Smoke and Dust," to which the engineers of the society are frequent contributors.

48 The work of the Hamburg society is as practical and effective scientific work as is done anywhere along these lines. The efficiency of the plants, under its control, is materially bettered and maintained, the owners derive a substantial economic gain and the smoke problem is solved without the aid of the usual unsatisfactory legislation.

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FOREIGN REVIEW

The aim of the Foreign Review is to present, within the available space, the main data contained in the articles indexed. Where possible, reference is made to English or American publications containing fuller information on the subject treated. Measures are given both in original units and their English equivalents. In many instances, engravings and tables are reproduced. Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer.

Aeronautics

INFLUENCE OF A LATERAL WIND ON THE EFFICIENCY OF PROPELLERS (*Influence d'un vent latéral sur le rendement des hélices*, P James. *L'Aérophile*, vol. 20, no. 17, p. 397, September 1, 1912. *e*). The experiments of Hiram Maxim and especially those of Riabushinski have shown that the quality of a propeller is improved considerably when it is immersed in a current of air inclined to the axis of the propeller. In the experiments of Riabushinski a propeller in calm air gave, at 35 r.p.s., a thrust of 0.036, with a quality of 0.08. In a current of air normal to the axis these quantities increased regularly with the velocity of the wind, and with a wind of 6 m (19.6 ft.) per sec., the propeller gave, at 28 r.p.s., a thrust of 0.082, with a quality 1.16, or a thrust 2.3 times, and a quality 14.5 times, better than in calm air. Since the power required to produce a certain thrust is inversely proportional to the square root of the quality, it follows that in the second case only one-quarter of the power was required to produce the same thrust as in the first case. It must be noted, however, that these results have been obtained with a stationary propeller and therefore apply directly only to the case of elevating propellers. The author believes that this may explain why propellers located behind the planes have a better efficiency; it is because they work in the air thrown downwards by the planes, and therefore making a certain angle with their axes.

BICYCLE WITH WINGS (*Das geflügelte Fahrrad*, Jos. Hofmann, *Prometheus*, vol. 23, no. 1190, p. 721, August 17, 1912. 4 pp., 4 figs. *t*). Theoretical investigation of conditions of *motorless flight*. The author believed that under certain conditions which he investigates in detail, such competitions as the one recently held in Paris may be of value.

RESEARCHES ON THE STABILITY OF AEROPLANES (Recherches sur la stabilité de l'aéroplane. *Bulletin de la classe des sciences, Académie Royale de Belgique*, J. Pacotte, 1912. No. 4, p. 292, 18 pp. *mt*). A theoretical investigation of the problem of *stability of aeroplanes*. The author's theory is based on the expression of the motions of an aeroplane by a system of nine differential equations. He introduces the notion of quasi-permanence, or a state tending towards permanence, and by this means limits the number of his equations to six, and further shows mathematically that a sufficiently steep ascent produces lateral instability, while a sufficient velocity produces lateral stability. The author investigates among other things the equation

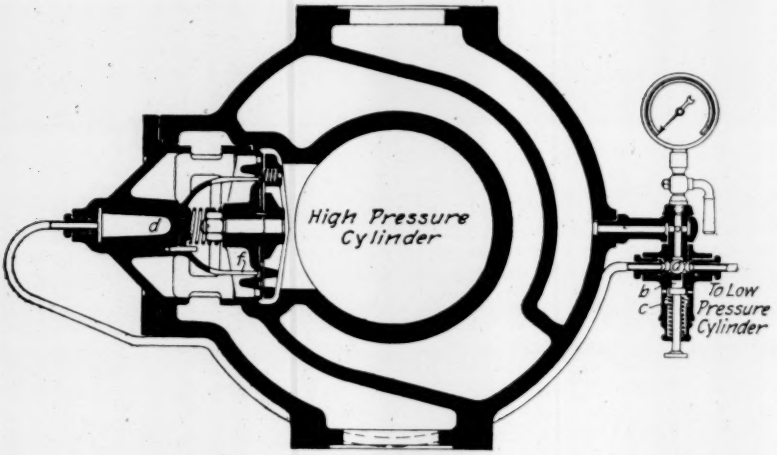

FIG. 1 AUTOMATIC PRESSURE REGULATION OF AIR COMPRESSOR

giving the criterion of stability; rolling, veering and pitching; necessary and sufficient conditions of stability. The whole subject is treated mathematically by elementary calculus in a very clear manner.

Air Machinery

A MODERN SWEDISH COMPRESSED AIR PLANT (*Eine neuzeitliche schwedische Druckluftanlage*, H. Eriksson, *Glückauf*, vol. 48, no. 34, p. 1368, August 24, 1912. 4 pp., 8 figs. *hd*). Description of the compressed air plant of the Striberg Company. Rogler-Egnell valves are used practically the same as the Hoerbiger-Rogler valves described in THE JOURNAL by W. Trinks.[1] The pressure regulator of the air compressor is shown in Fig. 1. In the regulator valve three passages come together at a single point, the upper passage communicating with the compressed air discharge piping, while the other two are connected with devices for closing the suction valves of the cylinders. At the point of intersection of the three passages is placed a steel ball *a* held air tight against the opening of the

[1] Reciprocating Blast-Furnace Blowing Engine, July, 1911, p. 775.

upper passage by the spiral spring *c* pressing against the piston *b*. When the pressure of the compressed air on the ball exceeds the pressure of the spring, the ball is forced downwards, and compressed air, flowing into the two side passages, is led through small diameter copper pipes to the suction valves of the high and low-pressure cylinders where it acts on the piston *d;* the grip *f* forces the valve plate from its seat and opens the valve; the low-pressure cylinder is now connected with the external air, and the high-pressure cylinder with the piping leading to the intermediate cooler, while the open suction valve connects both sides of the piston, so that the compressor takes only as much power as when driven at no-load. When the air pressure has gone down enough to permit the spring to drive the ball back against the opening of the upper passage, the admission of compressed air to the piston *d* is cut off, and the compressor starts to work again. By adjusting the spring *c*, the regulator may be regulated for any desired pressure.

Hydraulics

HYDRAULIC TURBINE GOVERNOR (*Régulateur de turbines hydrauliques.* Le Génie Civil, vol. 61, no. 16, p. 324, August 17, 1912. 1 p., 3 figs. *d*). Description of a new type of *hydraulic turbine governor* built by Briegleb and Hansen, of Gotha, Germany. The object of the introduction of the new type was to make a governor that would be entirely independent of the dimensions of the turbine or head of water, and one which could therefore be manufactured in series. This governor, as shown in Fig. 2A, consists of two sets of parts divided by the dotted line *xx:* above is the governor proper, the parts of which are subject to but slight stresses; below the mechanism guiding the action of the turbine vanes and therefore developing or transmitting considerable mechanical efforts; as a consequence they are much stronger than those of the first group. The rotary oil pump *P* drives the oil from the tank *C* into a pipe provided with the by-pass valve *S* which lets the oil back into the tank *C* as soon as the pressure of oil in the air tank *R* exceeds a certain predetermined maximum. From this pipe the oil passes to the slide-valve which admits it either above or below the piston *A*, in its turn directly connected with the flood-gates of the turbine which it opens in going down, and closes in going up. The governor proper (upper part of the figure, above the line *xx*) consists of a sensitive centrifugal governor *B* which acts on the lever *D* having its other end connected to the slide-valve *T*. The axis of rotation of the lever *D* is a knife-edge at the top of the vertical rod *L*, the length of which may be adjusted by means of the casing *G*. The rod *L* is connected with another rod *M* by a nipple *F* provided with a friction pulley rolling along the surface of the friction gear *E* which is held against it by the spring *s* and kept in continuous rotation transmitted to it from the centrifugal governor spindle by means of the bevel gear shown to the left of *s*. The rod *M*, which is a prolongation of *L*, is connected with the stem of the piston *A* by the slot *H* which permits regulating the ratio of vertical motions of the rod *M* and piston *A*.

If, when the speed of the turbine is normal, the load should happen to fall off, the speed tends to increase, and the governor balls deflect. The

lever D tips, with the end away from B going down, and forcing to its lower position the slide-valve T which opens to the oil under pressure a passage under the piston A, and sets the upper chamber of the cylinder in connection with the by-pass leading to the tank C. As a result the piston A rises, and thereby causes the flood-gates of the turbine to close. But at

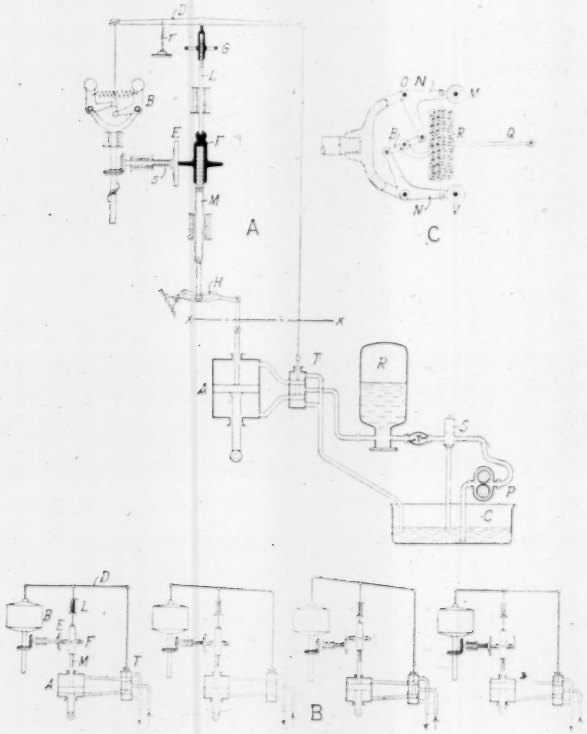

FIG. 2 WATER TURBINE GOVERNOR

the same time this motion of the piston is transmitted, by the rods M and L, to the central fulcrum of the lever D, and causes it to bring the slide-valve back to its neutral position. Fig. 2B shows the lever in its various positions. The article describes also how the turbine may be allowed to run at no-load at a speed slightly different from that at normal load, which is convenient in the case of turbines driving alternators in parallel. The centrifugal governor is shown in detail in Fig. 2C. Its great sensitiveness comes from the elimination of the sleeve and a certain number of levers and fulcra which increase the friction. The essential parts of the governor are two movable masses V subject to the action of the centrifugal force and set at the ends of the levers N, the other ends of these levers being interconnected by the feed rod B while the spring R provided with knife-edge contacts (to decrease friction to a minimum) tends to counteract

the tendency of the balls V to deflect. The spindle transmitting the motion of the masses V to the governor valves has its point of application at the center of the feed rod B.

ON THE SELECTION OF A TYPE OF WATER TURBINE IN DOUBTFUL CASES (*Über die Wahl der Wasserturbinengattung in zweifelhaften Fällen*, F. Leubner. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 31, p. 1244, August 3, 1912. 2½ pp. *p*). When from the turbine engineer's point of view either a Francis or a free-jet turbine may be applied, and the economic speed of either is convenient for direct driving the selected type of generator, the installation of a Francis turbine instead of a free-jet turbine may be recommended only when the water to be used is clean, and no prolonged underloading of the turbine is expected during the operation. The author shows that if the speed of rotation of a Francis turbine is low, while the head is high and the amount of water large, the Francis turbine works practically as a pressure turbine; there is a tendency of the water to work loose from the back side of the blades, and eddies are formed which help the gases contained in the water to become free. These gases attack the blades, which in a short time become worn, and sometimes full of holes. The advantages of the Francis turbine, viz. large power output and low first cost, must therefore never be solely considered when selecting a turbine type.

EXPERIMENTS IN MAKING VISIBLE FLOW PHENOMENA IN SUCTION PIPING (*Versuche über die Sichtbarmachung der Strömungserscheinungen im Saugrohr*, V. Kaplan. *Die Turbine*, vol. 8, no. 22, p. 403, August 20, 1912. 4 pp., 12 figs. *e*). Data from experiments at the German Technical High School in Brunn, Austria. Glass suction pipes were used for the observation of various processes in the suction pipe of water turbines, and instantaneous photographs taken at various moments.

Processes of Filling and Emptying of the Suction Pipes. Immediately following the admission of the water into the upper penstock, it flows through the suction pipe in a spiral, due to the rotation of the wheel at no-load. When the pipe is half full, the photograph shows the formation at the bottom of the pipe of an almost uniform mass of liquid, still containing however eddying air-bubbles. The limiting surface of the water core is indicated by white foam above which the water is still in spirals. A few seconds later the pipe is filled and, if it has an air tight connection with the wheel frame, it appears to be filled with a perfectly clear and transparent liquid. The process of emptying the pipe is similar to that of filling.

The problem of *making visible flow phenomena* in suction pipes was attacked in several ways of which only one proved to be satisfactory. Introduction of clay or chalk powder caused the water to lose some of its transparency without clearly indicating the flow phenomena owing to the small size of the particles. Solutions of coloring matter proved unsatisfactory because they formed a thorough mixture with the water, and the "colored line" became indistinguishable. A better method was found by introducing some liquid insoluble in water, e. g. oil or petroleum, but it

had the disadvantage of making the whole installation dirty. The author devised a method shown in Fig. 3 where *d* is a bunch of hemp introduced into the pipe on a piece of thin steel wire. The device indicated the flow phenomena very well when the hemp was not too long. The photographs in the article show the relation between speed of rotation, efficiency and direction of flow.

Internal Combustion Engines

NEW EXPERIMENTS ON THE COMBUSTION OF NITROGEN IN EXPLODING GAS MIXTURES (*Neue Versuche über die Stickstoffverbrennung in explodierenden Gasgemischen*, F. Häusser. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 29, p. 1157, July 20, 1912. 7 pp., 11 figs. *eA*). The *combustion of nitrogen at high temperatures*, proceeding according to the simple equation $N_2 + O_2 = 2NO$, is considered to be a purely thermal reaction depending exclusively on the temperature, but independent of the manner in which the temperature is produced. This was further proved by the experiments of Nernst, who passed air through externally heated platinum or iridium tubes and found that the production of oxide of nitrogen varies with temperature, or follows the equation of van't Hoff,

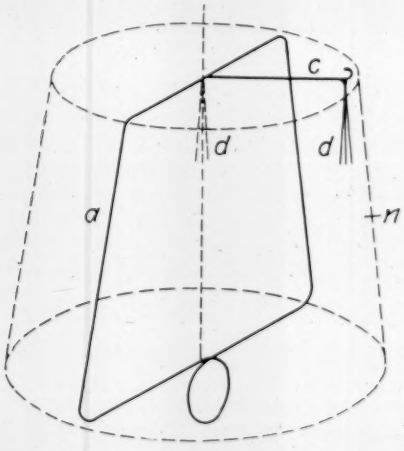

FIG. 3 APPARATUS FOR MAKING PHENOMENA OF FLOW IN WATER PIPES VISIBLE

$$\frac{d \ln K}{dT} = -\frac{q}{RT^2}$$

where q is the heat change due to reaction, R gas content, T absolute temperature of transformation, K constant of equilibrium, which is in this case determined by the equation

$$K = \frac{(NO)}{(N - 0.5NO)(O - 0.5NO)}$$

with NO contents of oxide of nitrogen, N and O contents of nitrogen and oxygen apart from what is contained in the oxide of nitrogen, all in per-

centage of volume. The integration of the van't Hoff equation gives, with decimal logarithms instead of natural,

$$\log K + \frac{9424}{T} = 1.07$$

which expresses also the maximum amount of oxide of nitrogen which can be formed in a gas mixture heated to T deg. and containing any amounts of nitrogen and oxygen, provided the formation of oxide of nitrogen is a purely thermal one, and the chemical equilibrium corresponds to the temperature T and is constantly maintained so.

In a series of experiments the author found that the contents of oxide of nitrogen varies with the size of the explosion chamber: while in his first experiments with small bombs the contents of oxide of nitrogen corresponded fairly well to what would be theoretically expected, in the later experiments with three times larger bombs the amount of oxide of nitrogen produced was found to be 86 to 116 per cent above what was theoretically expected. The author analyses the possible reasons for such a divergence, and ascribes the production of the excess of oxide of nitrogen to *photochemical effects*, a supposition partly supported by the experiments of Warburg and Berthelot with the formation of oxide of nitrogen by silent electric discharges in air. One of the interesting things shown by these experiments is that as the time during which the temperature remains at its maximum increases, the contents of oxide of nitrogen increases also, apparently according to a parabolic law. For the application of the above data to the fixation of atmospheric nitrogen see Engineering, September 6, 1912, p. 329.

Investigation of Carbureters of Motor Vehicles (*Studie über die Vergaser von Motorfahrzeugen*, A. Heller. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 27, p. 1075, July 6, 1912. 9 pp., 19 figs. *teA*). Surface carbureters are now antiquated although the fact that they produce actual evaporation of the fuel, and therefore deliver a mixture with no fuel present in liquid form, is an incentive to return to that type. The spray carbureter was first theoretically investigated by Krebs who proposed his well-known formulae for the velocities of the flow of fuel and air through a carbureter nozzle. He proved that as the speed of rotation of the engine and the partial vacuum in the carbureter increases, there must be additional passages open to the air in order to keep the ratio of mixture between fuel vapor and air invariable. Krebs embodied his ideas in his carbureter known as the Panhard and Levassor, and nearly all the carbureters in use are based on the Krebs law of additional air supply. Still the theoretical bases of the Krebs law are not correct because the flow of fuel through a nozzle of a carbureter does not follow the law $v_h = \sqrt{2gh}$ or $v_h = \sqrt{2g(h-h')}$, but, as was shown by Rummel and others, rather a law corresponding to the Poiseuille's law.[1]

The theory of internal friction leads to the assumption of proportionality between the pressure of frictional resistance and the first degree of velocity, or

$$p_r = cv$$

The lack of agreement between this formula and experimental data is usually

[1] For Poiseuille's law, see O. Reynolds, On the Motion of Water and the Law of Resistance in Parallel Channels, *Philosophical Transactions*, 1883, vol. 174, p. 935.

explained by the fact that while the theory of friction assumes a liquid jet of constant direction and close flow there are actually eddies of which the influence varies directly as the square of velocities. According to Reynolds the effect of eddies appears when the velocity exceeds a certain critical value (in c. g. s.)

$$v_k = 26 \frac{1}{d} \frac{\eta'}{s'}$$

where η' is the relative coefficient of internal friction, and s' the relative density of the fluid, both with respect to water of 10 deg. cent. (50 deg. fahr.). Reynolds found also that for the range above the critical point frictional resistance is proportional to the 1.7 power of the velocity. There is therefore in the region of the critical velocity a state of unstability with formation of eddies at the slightest disturbance. Further Grüneisen has shown that the value of the critical velocity

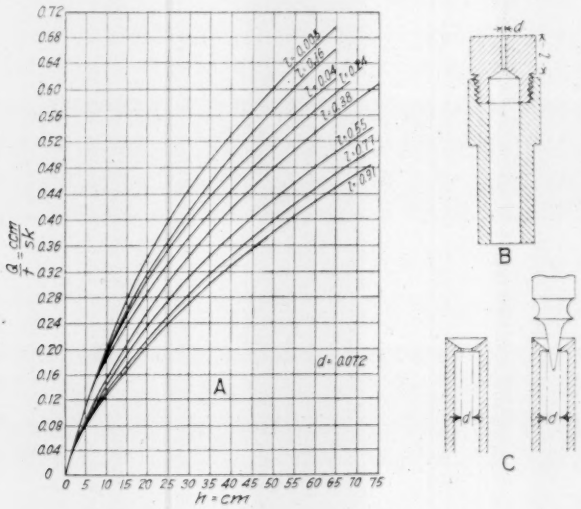

FIG. 4 *A* RUMMEL'S DATA ON FLOW OF WATER THROUGH CARBURETER NOZZLES; *B* AND *C* CARBURETER NOZZLES

at which values calculated from Poiseuille's formula differ from actual values by more than 0.1 per cent, is considerably lower, viz. (in c. g. s.):

$$v_k = 6.6 \times 10^{-6} \left(\frac{\eta'}{s'}\right) \frac{1}{d} \left(\frac{l}{d} - 4.5\right)^{2.08}$$

and therefore one cannot assume any longer the existence of a perfect proportionality between the first degree of the velocity of efflux and frictional resistance.

Rummel assumes for the frictional resistance a law of the form

$$\frac{p_r}{\gamma_B} = a_1 v_B + a_2 v_B^2$$

the basis for this assumption being that, theoretically, only the first 2 deg. of velocity can affect frictional resistance. He uses for the flow of fuel through the nozzle of a carbureter the usual law of flow of liquids

$$\frac{v_B^2}{2g}+h'+\frac{p_r}{\gamma_B}=h_o-h$$

where $h_o=0$ (atmospheric pressure). This finally gives him the following equations:

$$c_1 Q_B^2+C_2 Q_B=H$$

$$c_1=\left(\frac{1}{2g}+a_2\right)\frac{1}{F_B^2}$$

$$C_2=a_1\frac{1}{F_B}$$

where Q_B is the amount of fuel flowing in time t, and F_B the cross-section of the nozzle outlet. Subscript B refers to fuel (*Brennstoff*), and L to air (*Luft*). The values of c_1 and c_2 have to be determined for each case. Rummel made his experiments with water; Fig. 4*A* gives the results of his experiments on the flow of water through carbureter nozzles, made with a nozzle of variable length and 0.072 cm. (0.028 in.) in diameter. It is clear that frictional resistance has the greatest influence on the amount of fuel delivered by a carbureter nozzle. The author quotes some experiments on the flow of fuel through carbureter nozzles which show that the above equations are approximately correct (Cp. *Horseless Age*, August 19 and 26, 1908).

The flow of air through carbureter openings may be accepted to follow Zeuner's equation of velocity:

$$v_L=44.4\sqrt{T\left(1-\left[\frac{p_0}{p}\right]^{\frac{m-1}{n}}\right)}$$

where T is absolute initial temperature, p_o initial or external pressure, p pressure at the nozzle, m outflow exponent equal to 1.286. Since the variations of pressure in the case of flow through a nozzle are extremely slight, the formula may be simplified by neglecting the changes in specific volumes, this being permissible in such cases as was shown by the experiments of Durley.[1] This brings us to the Fligner formula for the quantity discharged; for the average temperature t this formula may be expressed in the form

$$\frac{Q_1}{t}=a\,F_L\sqrt{h}$$

The processes of flow of air and liquid fuel through the carbureter now appear to be approximately clear. The relation between the amount of fuel and air discharged and the partial vacuum H in the carbureter is such that it alone may sufficiently explain why the richness of the mixture increases with the increase in the speed of the motor.

In actual design an effort is made to keep the relation $z=\frac{Q_B}{Q_L}$ as much as possible constant for all values of H. This can be done in the following ways:

(*a*) The carbureter is adjusted usually by suitably adjusting the nozzle opening on the testing stand, for a given minimum partial vacuum so that at this partial vacuum z may have what is considered to be the best value (say 1:16), and then larger air passages are provided in order that, as

[1] Trans. Am. Soc. M. E., 1906, p. 193.

the speed of the motor rises, the minimum partial vacuum should not increase in proportion to the increase in the suction action of the motor. This is practically what the Krebs process does. Only automatic supplementary air valves may be used, since on the road the partial vacuum cannot be observed or measured so as to be adjusted by hand for various speeds. Though theoretically unattackable, the method of supplementary air admission in practice is far from being satisfactory. The adjusting and testing of carbureters, owing to the lack of proper methods of measurement, but seldom give sufficiently reliable results, and at the same time constitute a hindrance to mass production. It is inconvenient to have motors constructed on gages with no filing or scraping allowed, run for

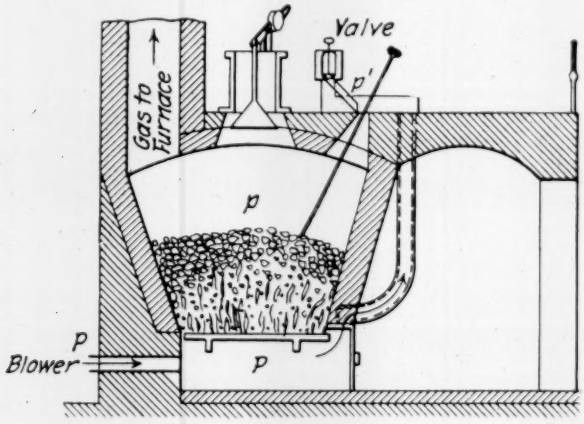

FIG. 5 GAS PRODUCER WITH PERNOT REGULATOR

hours on the testing stand only in order to have the carbureter adjusted. The curvilinear shape of the supplementary air passages cannot be copied so exactly in factory production as to require no subsequent filing, and it is just as little possible to produce springs in large amounts for the automatically acting air valves in which the tension, when the spring had been placed in the assembled carbureter, should vary in exact accordance with requirements established experimentally. The supplementary air method gives therefore machines which work well on the testing stand, but not in the average conditions of street work.

(*b*) The second method of maintaining the ratio of mixture $z = \frac{Q_B}{Q_L}$ constant is by having the air admission opening of constant cross-section, and varying the admission of fuel. This method is comparatively little used, because, especially with fine nozzle openings and in the hands of inexperienced drivers, it may lead to excessive consumption of fuel. If this method be applied at all, there ought to be a device for contracting the fuel passage when the motor begins to run at high speed; in most of the so-called carbureters with adjustable valves (*Registervergasern*),

however, when the partial vacuum increases, additional passages for fuel are opened (a full discussion of the Zenith carbureter follows).

The solution of the problem of constructing a carbureter that would work equally well at all partial vacua in the mixing chamber lies in finding a better form of nozzle. The usual carbureter nozzle is made generally with a cross-section like that shown in Fig. 4*B*, and to them applies Rummel's formula for the outflow

$$c_1 Q_B^2 + c_2 Q_B = H.$$

Should, however, the nozzle be designed that the simple parabolic law

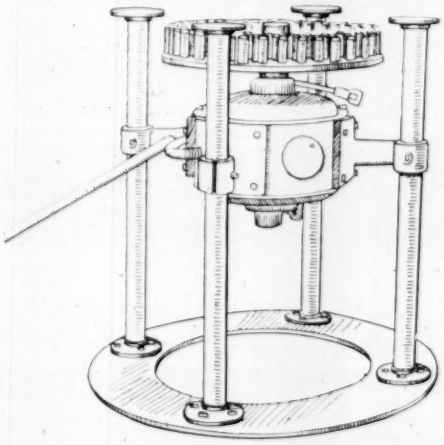

FIG. 6 OERLIKON CENTRIFUGAL SAND CLEANER

$cQ_B^2 = H$ might be applied to it, all the troubles of the present carbureter construction would be eliminated since the flow of air and liquid would follow the same law, and the ratio of mixture would be independent of the partial vacuum in the mixing chamber. The author proves experimentally that nozzles with a cross-section like that shown in Fig. 4C very closely approach this ideal condition in being nearly independent in their action of variations in partial vacuum. The peculiar characteristic of such nozzles is the extremely short length of their tubular part. In his work Rummel found that the regularity of the ratio of mixture may be improved by the use of very short nozzles. He, however, limited the length of nozzles to 1 mm (0.039 in.), but the shorter the nozzle, the less noticeable is the effect of capillarity which is the main cause of the flow of fuel differing from that of air at various partial vacua. It would therefore be reasonable to expect that in a nozzle with no capillarity at all, the flow would follow the parabolic law, just as in the case of flow through an opening in a thin wall. The article also contains data of tests on variation of pressure in the suction piping of a multi-cylinder engine.

REGULATOR FOR GAS PRODUCERS HAVING ARTIFICIAL DRAFT, C. PERNOT SYSTEM (*Régulateur de marche, système C. Pernot, pour gazogènes a tirage forcé*, A. Maillon. *La technique moderne*, vol. 5, no. 4, p. 137, August 15, 1912. ½ p., 1 fig. *d*). Description of a device intended to be used with gas producers having a draft produced by blowing in steam or air. If the pressure below the furnace be P (Fig. 5), there will be a pressure p, less than P, above the layer of coal. By means of a suitably arranged system of passages, a part of the fluid at the pressure P is conducted from under the furnace to an adjustable steam expander, whence it comes out with a pressure P' below P, but above p, and this fluid at a pressure P' is used for blowing in through the handholes while the attendant uses the slice-bar. If the pressure P' is conveniently chosen, it will prevent the gases at the pressure p from escaping from the furnace, which is important from a hygienic point of view. It is also important in regulating automatically the action of the producer. Suppose that the pressures P, p and P' have been so adjusted that there is no evolution of smoke or deleterious gases at the handholes. Should cracks appear in the fuel bed or should the bed become too thin, the ratio of pressures will change, and the furnaceman will have to see to it for his own convenience. The author states that this device was tried with satisfactory results with Siemens gas producers.

Machine Shops

NEW CENTRIFUGAL SAND CLEANER (*Neue Sandschleudermaschine*, H. Thieme. *Eisen-Zeitung*, vol. 33, no. 31, p. 556, August 3, 1912. 1 p., 3 figs. *d*). Description of a centrifugal sand cleaner lately placed on the market by the Oerlikon company. The sand is thrown from above into the hopper (Fig. 6), and falls upon a horizontal plate (Fig. 6) bolted to the end of the vertical shaft of the electromotor, and provided with three concentric rows of vertical staggered round bolts. The motor may work on direct current or alternating current, and take power from any lighting circuit. The sand is thrown by centrifugal force against the rows of bolts, forces its way through them and falls in a finely divided state in a circular heap on the ground ready for use. The iron particles stick in the middle of the plate and may be easily removed. The sand may be used for machine molding without further screening or other treatment. A 2-h.p. motor working at 960 r.p.m. can treat as much sand as two or three men can shovel into the hopper, or approximately 300 to 500 kg (660 to 1100 lb.) per minute. It does not require special foundations.

AUTOGENOUS WELDING OF COPPER AND COPPER ALLOYS (*Die autogene Schweissung von Kupfer und Kupferlegierungen. Acetylen*, vol. 15, no. 14, p. 125 (in the pagination of the appendix "*Autogene Metallbearbeitung.*" 5 pp., 5 figs. p). R. Amedéo, engineer of the Union de la Soudure Autogène in Paris has published a monograph on the above named subject in which he makes the following recommendations for the autogenous welding of copper: (*a*) the metal added (melt-bar) must have a melting point near that of the material welded; (*b*) it must absorb as little gas as possible, in order to produce a joint without porous places; (*c*) it must contain some deoxidizer capable of breaking up solutions of oxides; this

deoxidizer must be such as easily to penetrate throughout the metal and, after reducing the oxides, form such combinations which would not tend to remain in the molten metal; the amount of the deoxidizer present must be just large enough to deoxidize the metal melted by the welding flame, since an excess of the deoxidizer is apt to affect the mechanical qualities of the metal. Mr. Amedéo also points out that the use of electrolytic copper is entirely unsatisfactory, since it forms weak joints owing to the absence of some action preventing the formation of oxides. The application of welding fluxes is not sufficient to destroy copper oxides, because their action is superficial, and the oxides remain in solution in the molten metal.

The article seems to give preference to phosphorus-copper alloy for added metal, but contains also a discussion of copper alloys with aluminum, zink, tin and antimony.

INSTALLATION FOR AUTOMATIC PREPARATION OF CORE SAND IN THE FOUNDRY OF DE FRIES & CO., IN DÜSSELDORF, GERMANY (*Automatische Kernsandaufbereitungs-Anlage in der Giesserei von De Fries & Cie., Düsseldorf. Giesserei-Zeitung*, vol. 9, no. 16, p. 516, August 15, 1912. 2 pp., 4 figs. *d*). Pure Rhine gravel with from 3 to 5 per cent of very viscous oil is used for making cores for radiators and sectional boilers; as long as hand preparation and mixing were used, the distribution of oil throughout the core was not uniform and when the core was dried, cracks would appear. This difficulty was overcome when the mixing machine shown in Fig. 7 was introduced. The gravel in carloads is placed in the bunkers *a*, and thence conveyed by the conveyer *b* to the bin *c* provided at the bottom with a telescoping tube. The other end of the tube leads to a rotating wiper plate *d* which throws on to the tub mixing machine *e* all the gravel falling outside of the telescoping tube. The oil is brought to the mixing machine by a wheel with buckets which takes a definite amount of oil from the tank and delivers it to the machine through a channel opening into the mixer. From the mixer the sand is taken up by the conveyers *f* and *g* and delivered to the bin *h* located above the core benches.

Mechanics

THE GYROSTATIC EFFECT AND ITS APPLICATIONS (L'effet gyrostatique et ses applications, W. Bogaert. *Bulletin technique de l'association des ingénieurs sortis de l'École Polytechnique de Bruxelles*, vol. 10, Nos. 7, 8, 9, pp. 305, 393, 489. May-July 1912. 240 pp. 76 figs. *mtpA*). Owing to lack of space only the last part of this series of articles can be abstracted here. The author deals with systems in motion required to follow a certain path, but at the same time exposed to parasitic motions, the elimination of which constitutes the problem of stability. The gyrostat possesses the property of resisting, in proportion to the speed of its rotation, all efforts tending to change its direction of action (*désorientation*), and is therefore widely used in stabilizers, of which there may be two kinds: (*a*) the stabilizer of a normally stable system (pendulum, ship) which has only to keep the amplitude of oscillations within certain limits, and (*b*)

of a normally unstable system (bicycle, airship), where it must first establish a stable equilibrium, and then act as an oscillation absorber.

The stabilizer must therefore consist of (*a*) an apparatus sensitive to changes of orientation; (*b*) an element able to produce or absorb mechanical energy, and (*c*) a system of transmission between the two elements arranged in such a manner that the second should instantaneously develop the energy required to oppose any change of orientation indicated by the first. A gyrostat may be applied for stabilization in either of two ways: as an apparatus sensitive to changes of orientation, and directing the action

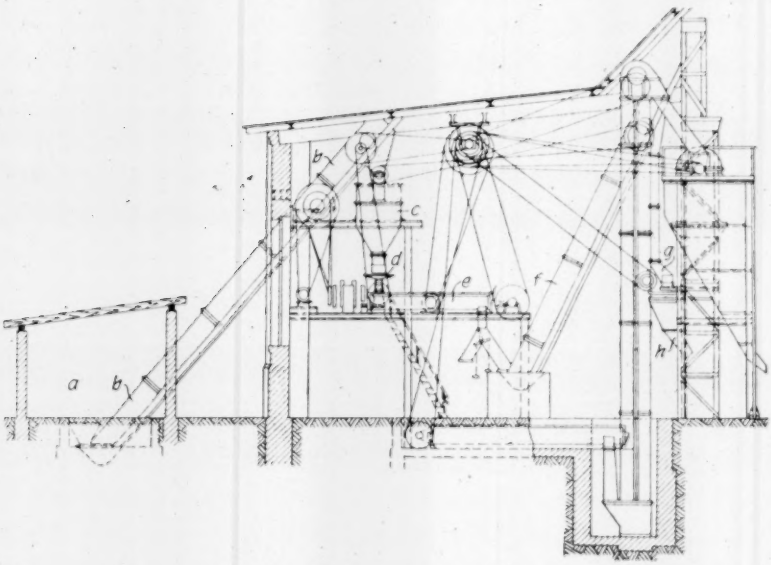

FIG. 7 HUMBOLDT ARRANGEMENT FOR AUTOMATIC PREPARATION OF CORE SAND

of a servo-motor, or it may also act as an intermediary between the source of energy and the stabilized system, and itself carry the whole effort of stabilization. As an apparatus indicating changes of orientation it is preferable to a simple pendulum or liquid level, since in order that the indications of the latter be instantaneous, they must have a long period which would make their deviations slight for small variations of position, while the gyrostat is sensitive not only to the change of orientation, but to its velocity as well.

The following rules must be applied when a gyrostat is used as stabilizers: (*a*) The axis of the tore must not be placed parallel to the axis of the oscillations to be reduced or prevented. (*b*) It must be given an average direction as far as possible normal to the axis of the undesirable oscillations. (*c*) If no invariable direction can be given to the axis of the tore, its variations must be limited in some way to prevent its deviation from normal to the axis of undesirable oscillations and thus from

exceeding a certain definite limit, e.g. 45 deg.; otherwise the gyrostat may take a position parallel to the axis of oscillations and become insensitive.

Among apparatus of the first class, i.e. with the gyrostat only indicating changes of orientation, the author describes in some detail the Obry and Howell torpedo apparatus, and the Regnard airship stabilizer. The theory of the stabilizers of the second class is treated in detail, the author establishing the following theorems for the case when the axis of the gyrostat does not deviate much from the average position normal to the axis of oscillations of the system. (Owing to lack of space, the proofs of the theorems have to be omitted): (*a*) A gyrostat taking part in a pendular motion has no effect on this motion if the position of its axis is invariable with reference to the system. (*b*) In the same case a gyrostat may introduce a stabilizing couple if the position of its axis is not invariable with reference to the system, i.e. if the gyrostat is capable of motions of precession; from this the author proves that an initially unstable pendulum may be rendered stable by the presence of a gyrostat having a kinetic moment greater than a certain definite value. (*c*) A gyrostat taking part in a pendular motion produces a stabilizing effect if the gyrostatic carriage, free as regards motions of precession, constitutes itself a compound pendulum. But the stabilizing action of the gyrostatic carriage acting as a pendulum is effective only when there is no friction.

As applications of the gyrostat as stabilizer, the author describes the ship stabilizer of Otto Schlick (including a discussion of the theory of ship rolling) and the monorail railway. The most valuable part of the article is the mathematical treatment of the theory of the gyrostat and its applications.

A list of books and articles on the gyrostat is appended to the article.

TORSIONAL OSCILLATIONS OF A ROD WITH FIXED AXIS OF ROTATION AND ANY DISTRIBUTION OF MASS SYMMETRICAL TO THE AXIS OF ROTATION, UNDER THE ACTION OF ANY HARMONIC FORCES (*Verdrehungs schwingungen eines Stabes mit fester Drehachse und beliebiger zur Drehachse symmetrischer Massenverteilung unter dem Einfluss beliebiger harmonischer Kräfte*, Professor Gümbel. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, nos. 26 and 27, pp. 1025 and 1085, 10 pp., 13 figs., *tA*). An interesting investigation passing in parts from the domain of mechanics to that of physics. The author shows that every elastic system rotatable about a definite axis and subject to the action of harmonic couples may be replaced, on condition of neglecting oscillations due to bending, by an elastic rod of zero mass and definite length with singular masses placed at certain distances from the axis of rotation. He investigates the motion of such a system under the action of periodic forces, and establishes a method for determining the variation in oscillations for each point at any frequency whatever. Three cases are fully investigated, and the connection between the spectrum and structure of matter discussed. The motion of an elastic system under the action of damping forces is investigated, and the application of the above theoretical considerations to technical problems discussed, particularly methods for preventing torsional oscillations. Such oscillations in the case of elastic systems of masses may be best prevented by avoiding the

use of periodically varying torques. If this cannot be done, the influence of periodically varying torques decreases as the difference between the free period of vibrations of the elastic system of masses and the frequency of the torques increases. In the case of technical applications of torsional oscillations, e.g. in ships, motor shafts, etc., the rule should be adhered to: to keep the frequency of the driving force at least 15 to 25 per cent below that of the free vibrations, when the latter are of the first order (with a point of inflection on the oscillation curve). If the oscillations are of higher orders, the oscillation curves for different frequencies have to be determined, and the frequency of the driving force to be selected in such a manner as to keep the variation in oscillations within permissible limits. If the problem is not to eliminate oscillations entirely, but only to keep free from them a certain part of a system, the curve of oscillations may be so modified by the addition of suitably placed masses or introduction of supplementary forces as to make the variation in oscillations for a certain point equal zero. To do this, each case must be investigated separately. The location of the point of application of the driving force is of considerable importance. If the force is applied at a point on the rod or shaft where the amplitude of torsional oscillations is zero, such a force can transmit no work to the rod, and (since a certain amount of work is necessary to overcome internal or external damping forces), will neither cause new oscillations, nor increase those already present. On the other hand, the driving force is all the more liable to produce oscillations, the greater the amplitude of vibrations at the point of application of the force.

Pumps

CONTRIBUTIONS TO THE THEORY OF PLUNGER PUMPS (*Beiträge zur Theorie der Kolbenpumpen*, Karl Mayer. *Die Fördertechnik*, vol. 5, no. 7, p. 148, July 1912 4½ pp., 1 fig. *tA*.). In double-acting twin pumps with equal working piston areas the coefficient of cyclic variation, the maximum velocity of the water column under suction or pressure, and the acceleration resistance may be reduced by keying on the cranks to an angle of coupling less than 90 deg. Another way in which this can be done in a double acting pump is to reduce in one pump the area of the piston on the side away from the crosshead, and to increase in equal proportion the piston area of the side of the connecting rod in the other pump so as to have the working piston area on the connecting rod side of one pump equal to that on the other side of the other pump, and vice versa. The author proceeds to show that the coefficient of cyclic variation, the maximum velocity of the water column under suction or pressure, and the acceleration resistance may be still more reduced by a combined application of the two methods above described. This may be used also in the case of twin differential pumps as well as for pumps with single-acting plunger pistons, provided in the latter case each pair of pistons is connected in the middle by a crosshead. He also shows that in the case of double-acting twin pumps with equal working piston areas the same may be done still more effectively by a suitable choice of the angle of coupling which may be both larger and smaller than 90 deg.

Fig. 8 gives the time-velocity diagram for an angle of coupling greater than 90 deg. The phases 2 and 4 are congruous throughout, while phases 1 and 3

are only related in a certain close way to the crank guide motion; in case of a rotating crank gear the compensation for error members of the second degree increase the maximum value of phase 1, and reduce that of phase 2. These compensations for error members of the second degree may, however, be neglected altogether since they are represented by a simple sinusoid of small amplitude and having practically zero value in the middle of the phase where the phases 1 and 3 reach their maximum. In a book published by him under the same title as this article, the author has shown that the maximum value of the phases 2 and 4 is

$$\max C_{r2} = \max C_{r4} = \frac{(F+f\cos 2\pi\sim x)\ (1+k\cos 2\pi\sim x)}{\sqrt{k^2+2k\cos 2\pi\sim x+1}} + \frac{kf\sin^2 2\pi\sim x}{\sqrt{k^2+2k\cos 2\pi\sim x+1}} \quad [1]$$

where the ratio of the working piston areas is expressed by $k=\frac{f}{F}$. Should the maximum values of the phases 1, 2, and 4 be equal to each other, the following equation holds good:

$$\frac{(F+f\cos 2\pi\sim x)\ (1+k\cos 2\pi\sim x)+kf\sin^2 2\pi\sim x}{\sqrt{k^2+2k\cos 2\pi\sim x+1}} = 2f\sin\pi\sim x+\lambda f\sin 2\pi\sim x \ldots [2]$$

After [2] has been divided by F and the numerator of the fraction simplified, it becomes clear that the later is only a square of the denominator, and equation [2] may be put in the simpler form:

$$\sqrt{k^2+1+2k\cos 2\pi\sim x} = 2k\sin\pi\sim x\ (1+\lambda\cos\pi\sim x) \ldots\ldots [3]$$

Should the maximum values of phases 1 and 3 be also equal, then

$$2F\sin\pi\sim x-\lambda F\sin 2\pi\sim x = 2f\sin\pi\sim x+\lambda f\sin 2\pi\sim x \ldots\ldots [4]$$

which may easily be reduced to

$$\cos\pi\sim x = \frac{1}{\lambda}\cdot\frac{1-k}{1+k} \ldots\ldots [5]$$

If the angular functions be eliminated from [3] and [5], the equation

$$(k-1)^2+\left(10\frac{1-k}{1+k}\right)^2 k = \left(\frac{4k}{1+k}\right)^2\left\{1-\left(5\frac{1-k}{1+k}\right)^2\right\} \ldots\ldots [6]$$

is obtained containing only one unknown k, which may be found by some method of approximation, e. g. the Newtonian process (it cannot be found directly, because the equation is of higher than fourth degree). The value

$$k=0.75545$$

satisfies the equation closely enough. When k has been found, the value of the angle of coupling for the minimum of the maximum velocity may be found from equation [5]. It is $x=91$ deg. 42 minutes.

The improvement of the maximum velocity in per cent is

$$\eta = 100\left\{1-\frac{2k\,|\sin\pi\sim x+\lambda\sin 2\pi\sim x|}{(\sqrt{2}+\lambda)\,\frac{k+1}{2}}\right\} \ldots\ldots [7]$$

and with $\lambda=\frac{r}{l}=\frac{1}{5}$ the value $\eta=14.89$ per cent is obtained.

The improvement (in per cent) when both methods are applied together is thus considerably greater than when either the method of selecting the best angle of coupling (the improvement in this case has been found to be only 6½ per cent),

or that of the best ratio of areas (improvement 10.52 per cent) *are applied separately.*

Coefficient of Variation of Raising of Water. The author proves mathematically that the coefficient of variation of the raising of water cannot be decreased by using the most suitable angle of coupling and ratio of sections on the basis of a comparison between the six intervals of variation in Fig. 8: the author speaks of six intervals of variation, because he proves that two intervals are nearly equal to zero respectively. He shows further that it is more advantageous to make the four largest intervals of variation equal to each other by equating the maxima values

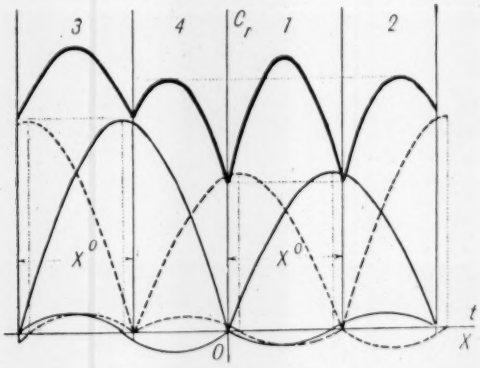

FIG. 8 TIME-VELOCITY DIAGRAM FOR PLUNGER PUMP WITH ANGLE OF COUPLING GREATER THAN 90 DEG.

in all four phases. With $\lambda = \frac{1}{5}$, the numerical value of the improvement in per cent, with the values of k and x as above, is

$$\eta = 100 \left\{ 1 - \frac{0.75545 \times 0.6023}{0.87772 \times 0.61421} \right\} = 15.48 \ldots\ldots\ldots\ldots [8]$$

The improvement in per cent of the coefficient of variation in the case of equality of maxima values in all the four phases is smaller only by a fraction of a per cent than in the case of the sole application of the method of the best angle of coupling with equal working areas of pistons, but the combined method has the great advantage of securing two minima values at the same time, viz.: minimum of the maximum velocity of the suction or pressure water column and minimum of coefficient of variation of raising of water.

Minimum of Resistance of Acceleration. The author uses in this connection practically the same method as in the preceding paragraph. He shows first that the acceleration pressure at the beginning of the fourth, or second, phase cannot be equal, with the crank mechanism ratio $\lambda = \frac{r}{l}$ within practicable limits, to the pressure at the end of the phase, since if they were equal, the following equation would hold:

$$f(1-\lambda)+F\cos 2\pi \sim x_s - F\lambda \cos 4\pi \sim x$$
$$= F(1+\lambda)+f\cos 2\pi \sim x + f\lambda \cos 4\pi \sim x = 0 \quad \ldots\ldots\ldots\ldots [9]$$

But this equation may be easily transformed into

$$1+\lambda\frac{1+k}{1-k} = \cos 2\pi \sim x - \lambda\frac{1+k}{1-k}\cos 4\pi \sim x \quad \ldots\ldots\ldots\ldots [10]$$

and that cannot be true as long as $\lambda = \frac{l}{r}$ is a rational fraction, because both $\cos 2\pi \sim x$ and $\cos 4\pi \sim x$ are less than unity and are negative, the angle of coupling being greater than 90 deg. The eight values of the maximum mass pressure cannot therefore be equal, and not more than six of them can be equal to each other. The author, as in the previous paragraph, proceeds to calculate the improvement, in per cent, in resistance to acceleration due to the use of the most favorable angle of coupling (x = 107 deg. 33 min.), and thus shows that the decrease of the resistance to acceleration due to the combined method is not larger than when either of these methods is applied separately. With a ratio $\lambda = \frac{r}{l} = \frac{1}{5}$, the acceleration pressure is the same as in a double-acting twin pump with working areas of pistons $\frac{f+F}{2}$ if the crank mechanism ratio is $\lambda = \frac{1}{10}$ and the angle of coupling of the pumps is 90 deg. The application of the combined method does not therefore affect the influence of the crank mechanism ratio on the resistance to acceleration any more than the application of either of the two methods separately. The acceleration pressure is nearly the same as when one of the two, the most favorable angle of coupling or ratio of areas, is selected. The article is to be continued.

EXPERIMENTS OF THE FIRM A. BORSIG IN TEGEL NEAR BERLIN WITH DUPLEX-SPRING VALVES IN HIGH-SPEED WATER PUMPS (*Versuche der Firma A. Borsig in Tegel bei Berlin mit Doppelfederventilen bei einer schnellgehenden Wasserpumpe*, O. Klepal. *Die Fördertechnik*, vol. 5, no. 8, 169, August 1912. 8 pp., 5 figs. cA). The author's fundamental contention is that for a steady and free-from-shocks operation an automatic valve must be loaded negatively, i. e. in the direction from the seat. The experiments were made with a differential pump having 60/85 mm (1.96/2.78 in.) diameter of the plunger piston, 200 mm (6.56 in.) stroke, with the suction and pressure air vessels arranged as shown in Fig. 9. The pump was driven by an electric motor provided with a device for varying the speed of rotation rapidly. Fig. 9 shows the construction of the suction and pressure valves which are both of the same design, viz. double-seat valves with two springs, of which one presses the valve on to its seat and the other lifts it up from the seat (in the rest of the article the pressure spring will be denoted by "+spring," and the raising spring by "—spring," the data referring to the first being denoted by subscripts d and those referring to the second by subscripts s.)

Owing to lack of space only data of the experiments which appeared to be the most characteristic have been reproduced here. The original numbers of the experiments have been retained to facilitate reference to the original.

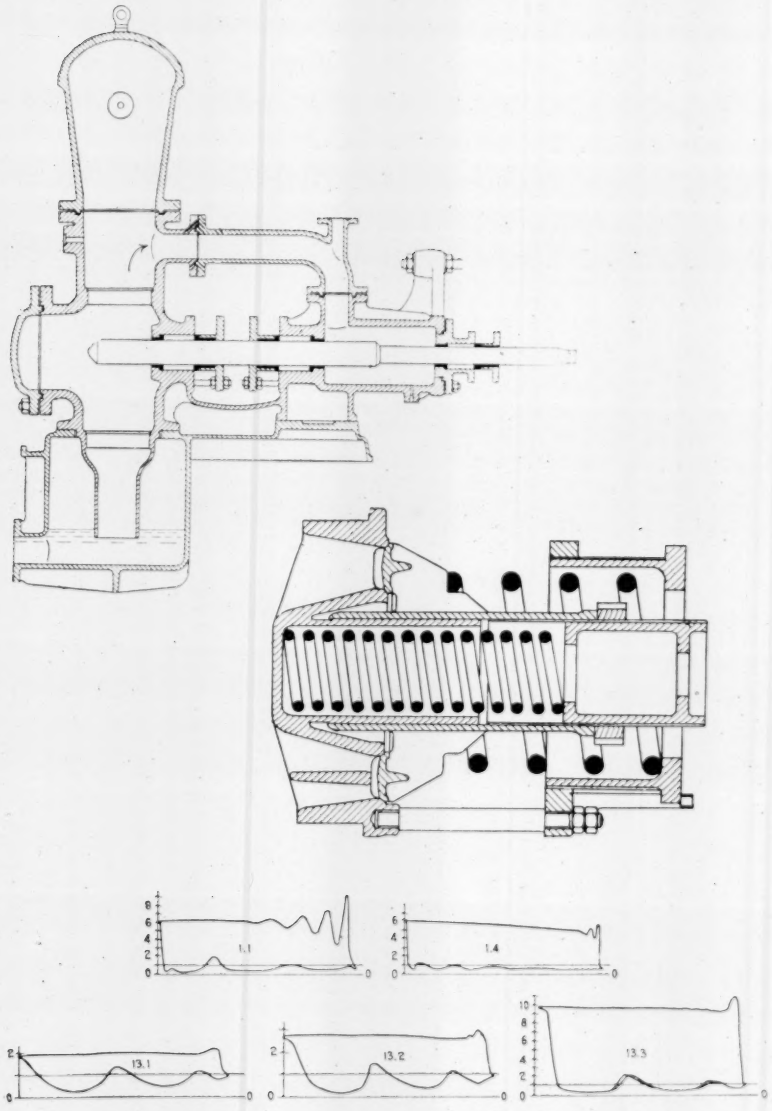

FIG. 9 HIGH-SPEED WATER PUMP WITH ITS VALVES, AND INDICATOR DIAGRAMS OF SAME

In experiment 1 both valves have negative loads, the pressure valve —7.79 kg. and the suction valve —4.95 kg. The two indicator diagrams are interesting in that they show how the indicator mass affects the form of the diagram, particularly that of the suction and pressure valves at the opening of the valve: the oscillations decrease with the decrease of the number of revolutions.

Experiment 11: Pressure valve negative load —7.86 kg. suction valve positive +30.8 kg; a Maihak indicator was used, with the result that the oscillations in the suction and pressure lines are smaller than when heavier indicators were used.

Experiment 13: Both valves are negatively loaded, —19.3 kg pressure valve, and —6.93 suction valve. This test shows clearer than any other one that it is possible to put a very large negative load on the valves without materially affecting the efficiency of the pump. At low pressures the suction line goes down, but improves at higher pressures, from 8 kg on. These diagrams show among other things that for each working pressure there must be a suitable load on the valves.

The highest speed was 330 r.p.m. at 9 atmospheres pressure. With suitably loaded valves the action was extremely regular, and only a very light metallic sound could be heard when standing behind the pump, but an observer standing at the crank bearing could not hear even that. Although the pump worked with 9 atmospheres pressure, it gave the impression of a machine working at no-load. The author states that other arrangements have been tried, but with none was it possible to reach a speed above 150 r.p.m., and the possibility of speeding the pump up to 330 r.p.m. with an arrangement as simple as this is certainly worth noticing. These tests have shown: that the pump refused to work whenever both valves were loaded positively; that the load on one valve affects the action of the other, and that the theoretical investigations which have treated the suction and pressure valves from the same standpoint, are incorrect; the speed of rotation as well as the working pressure affect materially the output of the pump and form of the indicator diagram.

Safety Appliances

ANNUAL REPORT OF THE TECHNICAL INSPECTOR OF SECTION IV OF THE "KNAPPSCHAFT-BERUFSGENOSSENSCHAFT" FOR THE YEAR 1911 (*Jahresbericht des technischen Aufsichtsbeamten der Sektion IV der Knappschafts-Berufsgenossenschaft für das Jahr 1911. Braunkohle*, vol. 11, no. 20, p. 313, August 16, 1912. 3 pp., *g*). The report contains, among other things, the description of the following noteworthy *safety appliances;* (*a*) apparatus for preventing bursting of flywheels on briquetting presses: when the flywheel speed exceeds a certain predetermined number of revolutions, a weight is so far projected by centrifugal force as to strike and release a lever, automatically closing the throttle valve and cutting off the steam admission; (*b*) under conveyor belts, just before the belt touches the roller with its hauling side, cleaning brushes are set rotating in a direction opposite to the motion of the belt. The rollers are therefore always clean, there is no further need to clean them while the machinery is in operation,

and the numerous cases of the cleaners having their hands or fingers drawn in between the belt and roller are avoided; (*c*) the foot operating lever of a compressed air hammer is protected by a sheet-iron plate which protects the hammer from being set into operation by a piece of metal falling from the anvil and striking it; a case like this occurred in the practice of the inspector, and the installation of such a cheap and simple protective device is recommended wherever the foot lever is used on steam and compressed air hammers; (*d*) device for protecting laborers cleaning manholes leading to drying ovens in briquette factories from being hurt by feeding of the fuel: a two-part iron cylinder is placed over the opening.

TABLE 1 DATA SHEET OF ZOELLY TURBINE TESTS AT DÜSSELDORF, GERMANY

	November 1, 1910			December 29, 1911		
No. of test	1	2	3	1	2	3
Duration of test, min	70	54	57	61	120	60
R.p.m.	3000	3000	3000	3000	3054	3036
Useful output at terminals, kw	277	541	781	1047	1044	547
Efficiency of generator	73.6	84.6	88.9	91.0	91	84.6
Pressure in front of turbine valve, atmosphere absolute	14.3	14.15	13.78	13.71	12.98	13.1
Temperature in front of turbine valve, deg. cent.	263	282	291.5	304	249	237
deg. fahr.	505.4	539.6	556	579	480	458.6
Pressure in turbine exhaust connecting branch, atmospheres absolute	0.043	0.052	0.065	0.079	0.083	0.057
Total consumption of steam per hour, kg.	2764	4346	5825	7199	7867	4770
lb.	6070	9561	12815	15838	17307	10495
Consumption of steam per useful kw-hr., kg.	9.91	8.03	7.46	6.87	7.51	8.71
lb.	21.8	17.8	16.5	15.2	16.6	19.4
Correction for steam temperature,[1] per cent.	−12.4	−9.5	−8.0	−6.9	−14.4	−16.1
Correction for vacuum,[2] per cent	+3.2	+3.2	+3.2	+3.2	+2.5	+2.5
Consumption of steam per kw-hr. at 350 deg. cent. (662 deg. fahr.) and 0.1 atmospheres absolute at full load, kg.	9.11	7.53	7.10	6.62	6.62	7.5
lb.	20.05	16.6	15.6	14.6	14.6	16.6

[1] 7 deg. cent. = 1 per cent.

[2] 1 per cent vacuum = 1.5 per cent steam consumption.

with a grate at the bottom nearly flush with the floor; the laborer stands on the grate, and is protected from falling coal by the cylinder.

Steam Engineering

RESULTS OF TESTS OF ZOELLY TURBINES (*Versuchsergebnisse an Zoelly-Turbinen*, L. Grabau. *Zeits. für das gesamte Turbinenwesen*, vol. 9, no. 23, p. 363, August 20, 1912. 1 p., 2 figs. *c*). At the plant of the Düsseldorf Tube Company, Düsseldorf, Germany, two *Zoelly turbines* were installed in August 1910 by Escher Wyss & Co., of Zurich, with a capacity of 1500 h.p. each, designed for a pressure of 12 atmospheres, superheat 350 deg. cent. (662 deg. fahr.), and 90 per cent vacuum at full load; each turbine has surface condensation with rotary air and circulation pumps, and

repeated cooling of cooling water. Each turbine drives a 550-volt alternating-current generator, with exciter.

Two series of tests were made: the first about 3 months after the two units went into operation and the second 13 months later. During that time the turbines worked about 4,000 hours, and delivered 1,450,000 kw-hrs. The purpose of the second series of tests was to find out whether or not there was any change in steam consumption during the above period: no such change has been found. Test data will be found in Table I. The consumption of steam of 6.87 kg. (15.2 lb.) per kw-hr. was at 92.1 per cent vacuum and 304 deg. cent. (579 deg. fahr.). Reduced to 350 deg. cent. (662 deg. fahr.) superheat, and 96 per cent vacuum, it would give a consumption of 6.01 kg. (13.2 lb.) per kw-hr. at full load of 1050 kw.

Modern Setting of Steam Boilers (*Moderne Dampfkesseleinmauerung*, Pradel. *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 35, p. 365, August 30, 1912. 3 pp., 7 figs. *dg*). A general description in a somewhat popular style of modern setting of steam boilers.

Plant for Coaling Boilers at the Mine Zollern II of the Gelsenkirch Mining Company (*Die Kesselbekohlanlage der Zeche Zollern II der Gelsenkirchener Bergwerks-A.-G.*, A Pietrkowski, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 29, p. 1164. July 20, 1912. 4 pp., 12 figs. *d*). Description of the *electric telpher line* used at the above plant *for conveying coal* from the washers to the boiler plant. The plant is an electric generating station capable of producing 12,500 kw. located at the coal mine of the company and supplying light and power to the surrounding communities and the city of Dortmund. About 100 tons of coal have to be delivered to the boilers daily. This is done mechanically by means of the described installation with an endless track lying at the elevation of the coal bunkers. The telpher cars are loaded either with nut coal from the washers or with rough coal; for the latter purpose they are lowered by special elevators to the loading openings located 6 m (19.68 ft.) below the level of the rails. The plant is designed to handle 35 tons per hour, and requires the attendance of only one man.

Resistance to Flow in the Distributing Valves of a Reciprocating Steam Engine (*Strömungswiderstände in den Steuerungventilen einer Kolbendampfmaschine*, E. Heinrich, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 30, p. 1191, July 27, 1912. 8 pp., 15 figs. *e*). Description of, and data from, experiments having for their purpose the determination, by means of indicator diagrams, of the *coefficient of flow in the admission and exhaust valves* of a single-cylinder condensing engine made at the engineering laboratory of the Royal Technical High School at Stuttgart, Germany. Only the final conclusions of this investigation can be given here. The coefficient of flow (or the ratio between the amount of steam actually flowing through a valve at any time during the process of flow to the instantaneous value of flow, or the amount of steam that would flow through the valve if its entire cross-section were utilized and if there were no resistances) increases with drop of pressure. The coefficient of flow considered as the ratio between the values stated above, but taken

from the beginning of the process of flow to the moment of observation (integrated value) may be taken for the admission valve to be approximately constant and equal to 0.55 for the admission curve beyond the dead point; for the exhaust valve it varies from 0.8 at the beginning to 0.62 at the end of the exhaust curve.

Strength of Materials and Materials of Construction

INVESTIGATION OF THE GERMAN SOCIETY OF BRIDGE AND IRON STRUCTURE BUILDERS OF IRON CONSTRUCTIONS USED IN BRIDGE BUILDING (*Untersuchungen des Vereines deutscher Brücken—und Eisenbaufabriken mit Eisenkonstruktionen für den Brückenbau*, B. Stock, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 28, p. 1101, July 13, 1912. 9 pp., 31 figs. *eA*). The

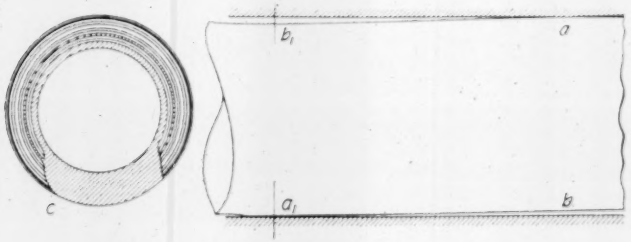

FIG. 10 YEAR-RINGS ON A LEAD PIPE

German Society of Bridge and Iron Structure Builders has been for many years carrying on a carefully planned series of investigations on *iron structures used in bridge building* through a special board formed of representatives of the Society, and the German ministries of Public Works, Education and Navy. The present article gives the main data of the third series of tests made under the direction of the board at the Royal Testing Laboratory at Gross-Lichterfelde West, viz.: (*a*) tension tests of riveted joints for the determination of the influence of the *process of riveting* (hand riveting, air riveting, toggle joint riveting with compressed air) on sliding resistance and breaking strength of the riveted joint; (*b*) tests on the *sliding resistance* in connecting plates with various forms of riveting; (*c*) tests on the influence of reduction of area on the tensile strength of *flat iron and angles;* (*d*) tests on bending of *angle webs.* All the rivets under tests were made without countersinking because it was found from previous experiments (1909) that *countersinking* in no way improves the qualities of the joint.

YEAR RINGS ON A LEAD PIPE (*Jahresringe an einem Bleirohr*, Kohlmeyer. *Chemiker-Zeitung*, vol. 36, no. 105, p. 993, August 31, 1912. 1 p., 1 fig. *d*). In making improvements in the water piping in the park of the Technical High School at Charlottenburg there was uncovered a lead pipe which had lain in the ground for 24 years at a depth of about 1 m (say 3.3 ft.). Part of the pipe lay in the earth unprotected, while one of the ends was imbedded in a cement foundation. The part which lay in the earth was found to be in a practically perfect state of preservation, but the part

imbedded in cement underwent a rather peculiar transformation. As shown in Fig. 10B, the part of the pipe in the cement was slightly bent, and the contact with the cement was not uniform. It was found that just where the pipe entered into the cement, the lead was transformed into a compact, stonelike, yellow-red mass, and when this was knocked off near a part of the pipe where the lead was not affected, the latter had a scarred surface with silver white lustre, unusual for lead, which rapidly changed to a dull gray in air. The pipe was sawed through at b_1a_1 and

FIG. 11 LUBECK DIVING APPARATUS

the section polished (Fig. 10A). The pipe was 18 mm (0.7 in.) in diameter, with the walls 5 mm (0.196 in.) thick; at b_1 the yellow-red mass was 4 mm (0.15 in.) thick, but its thickness decreased gradually toward a_1. The most remarkable fact, however, was that on closer investigation it was found that the transformed mass consisted of concentric alternating yellow and red layers, 24 red and 23 yellow, or practically *one pair for each year the pipe lay in the cement*: the layers of one color were evidently formed during the wet, and those of the other color during the dry part of the year. An analysis proved that the colored substance was lead oxide, the yellow part being PbO, and the red Pb_3O_4. The fact that only the part lying in cement was so affected, appears to show that this peculiar oxidation of lead was due to the action of either the cement or cement water. The author undertook

a series of tests in this connection, of which he promises to publish the results in a few years.

Miscellanea

SOMETHING ABOUT DIVING (*Etwas von der Taucherei. Prometheus*, vol. 23, no. 1188 and 1189, p. 689 and 713, August 3 and 10, 1912. 8 pp., 12 figs. *hd*). Description of various types of diving apparatus, from the Smeaton diving bell to that used at the present time. An interesting type is the Lubeck *diving apparatus without outside air supply*. The diver is provided with a cylinder filled with compressed oxygen besides the usual compressed air cylinder. When the apparatus is in operation, the oxygen flows through a pressure-reducing valve into a nozzle, sucks the used air through a hose (Fig. 11 at the left), and forces it through a vessel filled with anhydrous potash which absorbs almost entirely the moisture and carbon dioxide contained in the air; it returns the air through the hose at the right into the diver's helm supplied with oxygen and fit for breathing. The air supply of the apparatus is in continual circulation, and the oxygen used up is replaced by new. The diver receives 2 liters (0.76 cu. ft.) of oxygen per minute, and 60 liters (22.8 cu. ft.) of air. The oxygen and potash tanks are so proportioned that the diver is provided for two hours' stay under water; a lever at the belt allows him to regulate the consumption of air and oxygen. The inside pressure is regulated by an automatic valve placed in front at the shoulder which lets out the air whenever the inside pressure exceeds the water pressure by a certain small amount. The diver can decrease the inside pressure or blow out his suit by the air from the compressed air tank as he wishes, a considerable convenience in work at varying depths. The absence of an air hose is said to make this apparatus especially convenient for deep working.

The second part of the article is of a historical character, and contains descriptions of diving bells of the older types.

INDUSTRIAL MECHANICAL TREATMENT OF STONE (*Le travail mécanique de la pierre dans l'industrie*, J. Escard. *Revue de mécanique*, vol. 31, no. 1, p. 5, July 31, 1912. 29 pp., 44 figs. *d*). First of a series of articles on machinery used in the working of stone. The present article describes in detail various *saws for cutting stone:* reciprocating and band saws, metal and diamond, with fixed and movable teeth.

SOME SPECIAL TESTS OF AUTOMOBILE CARRIAGES (*Quelques essais spéciaux sur les voitures automobiles*, Boyer-Guillon. *Revue de mécanique*, vol. 31, no. 1, p. 30, July 31, 1912. 25 pp., 15 figs. *eA*). Description of a very simple and convenient apparatus used for testing automobiles by the testing laboratory of the Conservatoire National des Arts et Métiers, and of tests on automobile suspensions and shock-absorbers.

Supplementary References

Miller for Torpedo Propeller Blades (*Foreign Review*, September 1912, p. 1390). Detailed description in *American Machinist*, vol. 37, no. 11, p. 457, September 12, 1912.

The Gyrostatic Effect and its Applications (p. 1571). Cp. Herbert Chatley, *Practical Gyrostatic Balancing*, London, 1912, particularly p. 16 ff.

GAS POWER SECTION

PRELIMINARY REPORT OF LITERATURE COMMITTEE

(XXI)

ARTICLES IN PERIODICALS[1]

ABWÄRMEVERWERTUNG BEI VERBRENNUNGSKRAFTMASCHINEN, K. Kutzbach. *Stahl und Eisen*, July 11, 1912. 2 ⅔ pp., 4 figs., 2 curves. *dp.*

Utilization of waste heat from internal-combustion engines.

ELYRIA OIL ENGINE. *Power*, August 27, 1912. 1⅓ pp., 3 figs. *d.*

Describes a two-stroke cycle oil engine of the Diesel type.

ENGINE, 400-B.H.P. TWO-CYCLE GAS. *Engineering*, August 30, 1912. 2 pp., 6 figs. *d.*

Describes a single-cylinder engine without any side shafts or eccentric gearing.

EXHAUST GAS TURBINE, AN. *The Engineer* (London), August 23, 1912. 2 pp., 4 figs., 1 table. *depA.*

Gas turbine connected to exhaust of petrol motor; theory and design of turbine by Mr. Montague Churchill-Shann, London.

EUROPEAN GAS ENGINE PLANTS, SOME RECENT, A. G. Lescarboura. *Power*, August 27, 1912. 1⅓ pp., 3 figs. *d.*

INTERNAL-COMBUSTION ENGINES, AMERICAN PRACTICE IN RATING, Parts 1, 2, 3, 4. T. C. Ulbricht and C. E. Torrance, Jr. *Power*, July 23, 30, August 6 and 13, 1912. 8½ pp., 9 tables, 36 curves. *em.*

This series of articles is the result of a thesis investigation made under the direction of Profs. C. F. Hirshfield and H. Diederiches of Cornell University.

JUNKERS OIL ENGINE, THE, J. B. Baker. *The Iron Age*, August 1, 1912. 2 pp., 8 figs., 1 curve.

Valveless engine with two pistons per cylinder for utilizing cheapest grades of fuel.

MARINE OIL ENGINE, 10-H.P. "MONOBLOC." *Engineering*, August 16, 1912. 2 pp., 3 figs. *d.*

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[1] Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *h* historical; *m* mathematical; *p* practical. A rating is occasionally given by the reviewer, as *A*, *B*, *C*. The first installment was given in The Journal for May 1910.

MOTEURS À COMBUSTION, ÉTUDE THEORIQUE DU RENDEMENT THERMIQUE DES, T. Mancy. *Revue de mécanique*, June 20, 1912. 3 pp., 5 curves.

Theoretical study of thermal operation in internal-combustion engines.

MOTOR SHIP MONTE PENEDO, THE. *The Engineer* (London), August 30, 1912. 1 p., 2 figs., 1 table. *dpB.*

The ship, which is 350 ft. long by 50 ft. beam, is equipped with two Sulzer two-cycle Diesel engines; 1700 combined h.p. at 160 r.p.m.; 0.46 lb. Texas oil consumption per b.h.p.

NECROLOGY

E. HARRISON SYMINGTON

E. Harrison Symington was born in Baltimore, Md., March 1, 1879, and was educated in the public schools there and in the technical training school at Bethlehem, Pa. He entered Lehigh University in the class of 1898 and was graduated with the degree of M.E. His first experience was in the shop of the Lehigh Valley Railroad, with which company he remained till he was made engineer of tests, serving in that capacity for one year. In 1897 he entered the employ of Kilborne & Jacob Manufacturing Company of Columbus, Ohio, where he was in charge of the engineering and shop practice. He resigned this position to become manager of western sales of the T. H. Symington Company, and subsequently of their foreign business. When the concern opened its new malleable foundry in Rochester, N. Y., 1909, he was made works sales manager. His last years were spent as mechanical expert for the company, especially devising improved methods for draft gear designs for freight equipment.

Mr. Symington died in Baltimore, September 5, 1912.

FREDERICK J. MAYER

Frederick J. Mayer was born in Bremerhaven, Germany, July 4, 1853. He received his technical education at the University of Stuttgart, and came to this country in 1870, after having seen active service in the Franco-Prussian War. He engaged with the Bartlett Hayward Company of Baltimore, Md., as draftsman, and while with them, as chief engineer, he designed, erected and operated many of the large gas plants constructed by the company. Among them may be mentioned the Consumers' Gas Fuel and Light Company of Chicago, in 1883, and the Nassau Gas Company's water gas plant, Brooklyn, in 1891. In 1906 Mr. Mayer severed his connection with the Bartlett Hayward Company and organized and established the Didier-March Company in the United States, taking the position of general manager.